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GENERAL INDEX

THE DESIGN AND PROGRESS OF THE FLOATING-FRAME REDUCTION GEAR

By John H. Macalpine*

The gears of which I am about to speak all belong to the class of toothed gearing. They are called reduction gears because, almost invariably, a pinion actuates a larger gear-wheel and thus from the higher speed of the motor which drives the pinion there is produced a reduced speed of rotation more suitable for the purpose in hand. There are also hydraulic and electrical reduction gears to which I do not propose to refer further. As the title indicates, this paper deals principally with floating-frame gears of which the Westinghouse Machine Company now has under construction over 2 900 000 horsepower.

No doubt the transmission of power was the object of the invention of toothed gearing but this would usually be accompanied by change of velocity of rotation. It is a safe surmise that the object was, most frequently, to exert a greater force by means of a weaker, and if so, reduction gears date back at least to the time of Archimedes in the third century B. C. They are thus more venerable than the steam turbine—that motor with which they are so commonly associated to-day—which, as is well known, dates from Hero's reaction turbine in the second century B. C.

It is not proposed to-night to study ancient history but it is most interesting in passing to note that a machine, used to turn a spit, which suggests both the gas turbine and the screw propeller, geared, was invented in 1500 A. D. by that famous artist and engineer, Leonardo da Vinci. But the invention of the hydraulic turbine and screw propeller seems to "date from the beginning of the 15th century and it is ascribed, very strangely, to a 'pope of Rome'. The name of this wearer of the tiara is not definitely known." Illustrating this, a drawing dating from about 1575 seems to exist which shows the form of turbine reinvented by Poncelet in 1826. The power is transmitted through

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two sets of gears, one pair spur and the other bevel gears. The teeth are round pins, a form of tooth still to be found in some clocks.*

The next geared turbine we find is that of Giovanni Branca, 1629, in which a pinion is mounted on the vertical spindle of an impulse turbine which actuates a horizontal shaft through a gear-wheel. This turbine, as figured, was no advance on that of Hero (which, in a test by Sir Charles A. Parsons, some years ago, gave very fair results) and the design of the gearing, with its pin teeth, in no way suggests the refinements of a modern reduction gear.

Invention relating to the geared turbine seems to have lain dormant for over 250 years. But in 1889, Dr. Gustaf De Laval, discarding an earlier form of turbine he had used, introduced his geared turbine (Fig. 1), which quickly became well known and

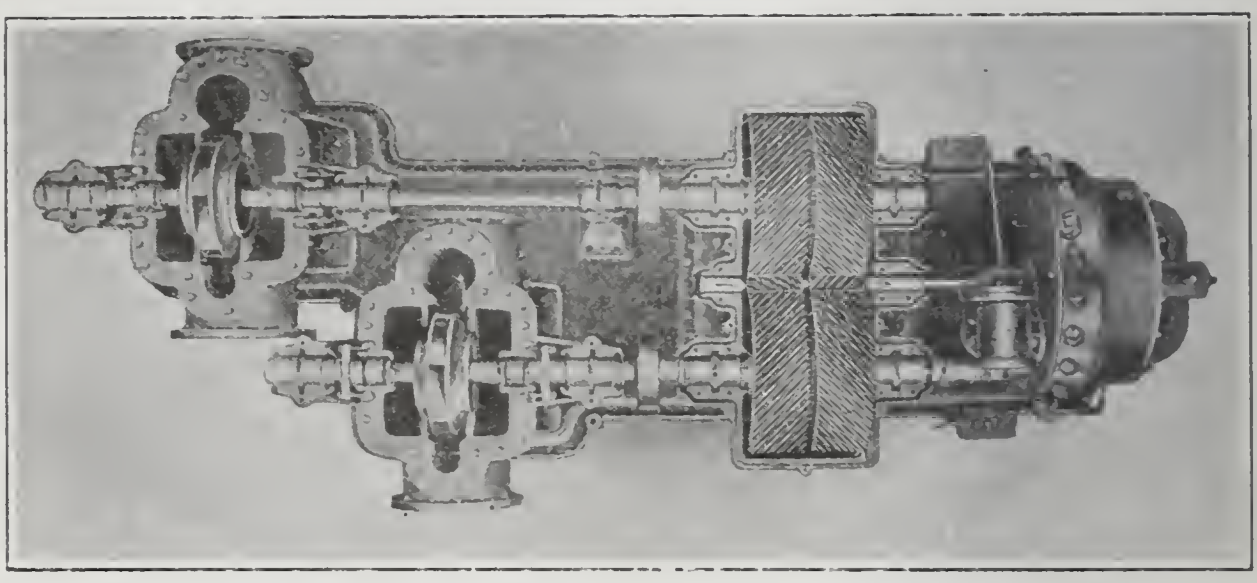

Fig. 1.

has met with considerable success up to about 200 horsepower. For good economy this single-stage turbine must run very fast and therefore the ratio of gear to pinion diameter must be large in order that the speed of rotation of the large gear may be sufficiently reduced for driving pumps, blowers, and so forth. Thus the pinion is made very small in diameter so that the gear-wheel diameter may be kept within reasonable limits. To give sufficient tooth contact this small pinion must be made as long as prac-

*For this and Leonardo da Vinci's inventions see Scientific American Supplement, February 17, 1906, page 25188.

ticable. This introduces two difficulties of great importance: 1. A slender pinion is liable to cross-bending between the bearings. 2. The torsional yield of the pinion becomes important. Both these distortions cause bad distribution of tooth pressure. The torsional error is in no way counteracted in the De Laval gear, and would, alone, limit the average pressure which can be applied per inch of tooth. The cross-bending is prevented by placing the pinion, in all but turbines of small power, between two gear-wheels. It thus pushes up on one side and down on the other. If the powers transmitted by the gear-wheels are equal there is no cross-bending of the pinion, but it is not possible to maintain this equality at all times and then serious cross-bending may occur. Thus this is never a safe way of eliminating the cross-bending error in any gear design. More important than either of these, in any gear whatever which has all the pinion and gear bearings cast in one solid housing, the slightest warping of this casting, or other causes, will put the pinion out of line and cause highly localized and intensified tooth pressures, especially if the pinion is long. Where there are two gear-wheels, as here, this danger is intensified. Even if set true at first, misalignment is sure to occur during the life of the gear. These are, I believe, the points in the design which have confined the De Laval gear to small powers.

Up to this time the geared turbine had not been applied to ship propulsion. Gearing had commonly been used in the early days of the screw propeller but then it was fitted to speed up the slow revolutions of the reciprocating engines of that day to a speed suitable to the propeller. When I was in Robert Napier & Sons, Glasgow, in the eighties, drawings of geared marine engines were still on file and I regret that I did not examine them closely and make a record. But a great departure must now be noticed. In 1884 Sir Charles A. Parsons took his first turbine patent. These turbines he applied on land with increasing success and later fitted in the steam yacht *Turbinia*. This ship so far exceeded in speed the fastest then afloat that naval architects all over the world were amazed. Gradually the direct-connected turbine was introduced and very many ships of high and even moderate speed have been fitted thus. The turbines of the *Tur-*

binia were directly connected to the propellers but it must have been obvious to Sir Charles, as it was later to any who gave the subject a moment's serious thought, that success with the direct-connected turbine was possible only in a very fast ship. Obviously gearing would overcome this difficulty, as it would reduce the high speed necessary to an economical turbine to the moderate revolutions per minute requisite for an efficient propeller working in a dense medium—water. In 1897 the Parsons Marine Steam Turbine Company fitted a geared turbine of 10 horsepower in a twin-screw launch. There was a pinion between two gear-wheels one of which was mounted on each propeller shaft. But in 1901 after, apparently, numerous experiments we find that Sir Charles had concluded that gears, except for very small powers, were wholly impracticable.

It was at this stage that the problem was taken up by Admiral Melville and myself, the result being the invention of the floating frame—a stiff frame which carries the pinion bearings and automatically adjusts itself to give a good distribution of tooth pressure. This will be fully illustrated.

Soon after the late Admiral George W. Melville retired from the post of Engineer-in-Chief of the United States Navy he and I determined to start in Philadelphia as consulting engineers. But before this could be done Mr. George Westinghouse told Admiral Melville that he was about to start a marine turbine department of the Westinghouse Machine Company and made him a handsome offer to take charge of it. This led to considerable discussion in which we told Mr. Westinghouse that we were very doubtful of the superiority of the marine turbine—or, as we would now say, the direct-connected turbine—to the reciprocating engine. It was applicable with fair success only to very fast ships; and, even at full speed of ship, the turbines ran much under their most economical speed, and the propeller ran too fast for good efficiency. That is, even at full speed there was a compromise, and usually not a good compromise, between the best speed for the turbine—a high speed motor—and that for the propeller. At reduced speed of ship the efficiency of the turbine could not fail to be seriously lowered while that of the recipro-

cating engine is usually increased. The fact that results as to coal consumption of these early turbine ships had been carefully concealed was a sure indication that performance had been unsatisfactory, and the coal consumption far above the sanguine hopes which had been formed. The slow running turbines had to be made large in diameter and very long which made them exceedingly heavy and subject to difficulties of expansion. This latter was borne out by many stories of blade stripping which leaked out. The only unquestioned advance which had been made was that higher speeds of ship became possible, and this increase was very remarkable.

Mr. Westinghouse had to go to Europe early in 1904 and as a result of our discussions he asked us to accompany him that we might investigate the status of the marine turbine. In a tour of three months duration, besides visiting England, Scotland and Ireland, we went to Holland, Germany, Switzerland and France. In May 1904, soon after our return, we handed Mr. Westinghouse a report which showed cumulative evidence of the truth of the tentative criticisms we had previously made. This report deterred him from forming the proposed marine turbine department, and he expressed himself later as being convinced that it would have been far from a success. He had the report very beautifully printed, illustrated by cuts which we obtained from turbine and ship builders in the countries we had visited. It was very widely distributed and while it was received with interest by many it did not escape adverse criticism by some who were deeply interested in the direct-connected turbine. It was reprinted, again for private distribution, in 1909 without any change, in a book entitled "Broadening the Field of the Marine Turbine" which gave an account of the first Melville and Macalpine reduction gear and its trials to 6000 horsepower, of which I will have to speak immediately. When the geared marine turbine began to be adopted years later, those who had criticized our 1904 report took exactly the same position as we had stated, and that position is now taken by all turbine engineers.

While we were abroad we told Mr. Westinghouse that the only hope of applying the turbine with complete success to ships, fast or slow, lay in the adoption of gearing. But one of the diffi-

culties which I have already noted in connection with the gear of the De Laval turbine presented itself—to keep the whole apparatus as compact as possible, so that it could be adapted to the limited space available in a ship, the pinion and gear diameters had to be kept down. In a ship compactness is also specially important in order to save weight, as every additional ton adds one ton dead weight which has to be unprofitably driven through the water. This small diameter of the pinion necessitated making it as long as practicable so as to carry the maximum horsepower. Putting aside for the moment the errors due to cross-bending and torsion there remained the one of outstanding importance—the slightest error of alignment of the pinion and gear axes will localize the tooth pressures to a short portion of the long tooth face. As already stated, even if correctly adjusted, the slightest warping of the housing, or other changes, which are sure to occur, would completely upset the uniform distribution of tooth pressure and seriously lower the maximum load which could be carried. That this conclusion was correct I will show later.

We were so deeply impressed by this difficulty that in our 1904 report we did not refer to gearing, but we appreciated so highly the enormous value of the marine problem presented that we stated in the conclusion, "If one could devise a means of reconciling, in a practical manner, the necessary high speed of revolution of the turbine with the comparatively low rate of revolution required by an efficient propeller, the problem would be solved and the turbine would practically wipe out the reciprocating engine for the propulsion of ships". This evolution is now taking place very rapidly and will be complete before many years.

After much discussion between Admiral Melville and myself, and before the solution of the problem by means of the floating frame had suggested itself, we made a design of an ordinary pinion and gear, the bearings being part of the housing. Such a gear I will refer to as a "rigid gear" since there is no automatic yielding or adjustment of the bearings. The teeth were involute and helical, one half being a right-hand helix and the other half a left-hand. The reasons for this will appear later. The only feature of the design which need be referred to is that a middle bearing was interposed between the right- and left-hand helixes

of the pinion, making with the two end bearings a three-bearing pinion. This, as can readily be shown, reduces the cross-bending error to about 1/20 of what it would have been with a two-bearing pinion.

Mr. Westinghouse proposed to build and test this gear, but while estimates were being sought for building the gear and cutting the teeth there was ample time to critically examine the many important questions involved in the design. The results of this examination were set down in a document dated February 26, 1906, called "Comparison of Reduction Gear of 2500 Horse Power with Gear of 300 Horse Power De Laval Turbine". This

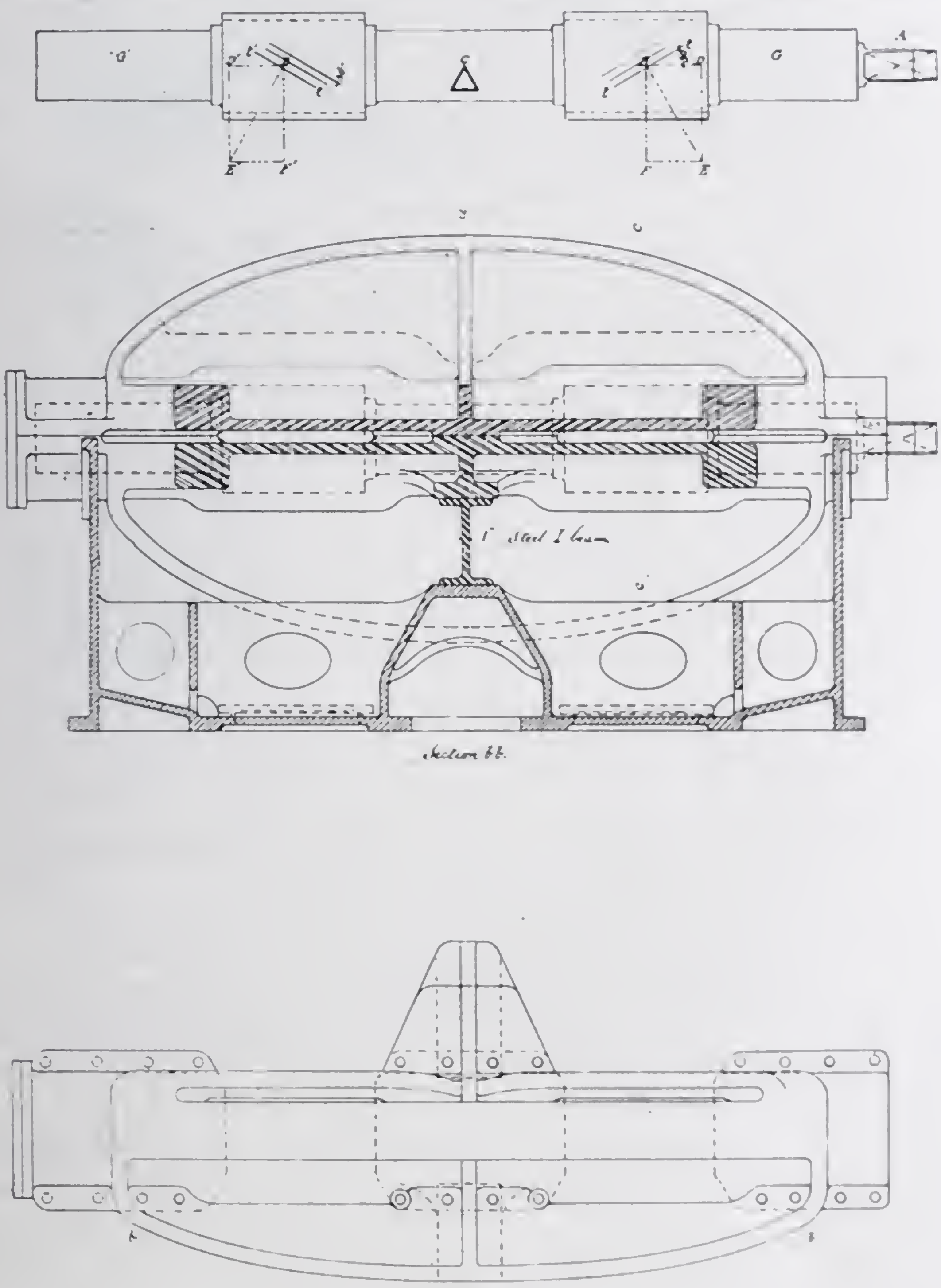

Fig. 2A.

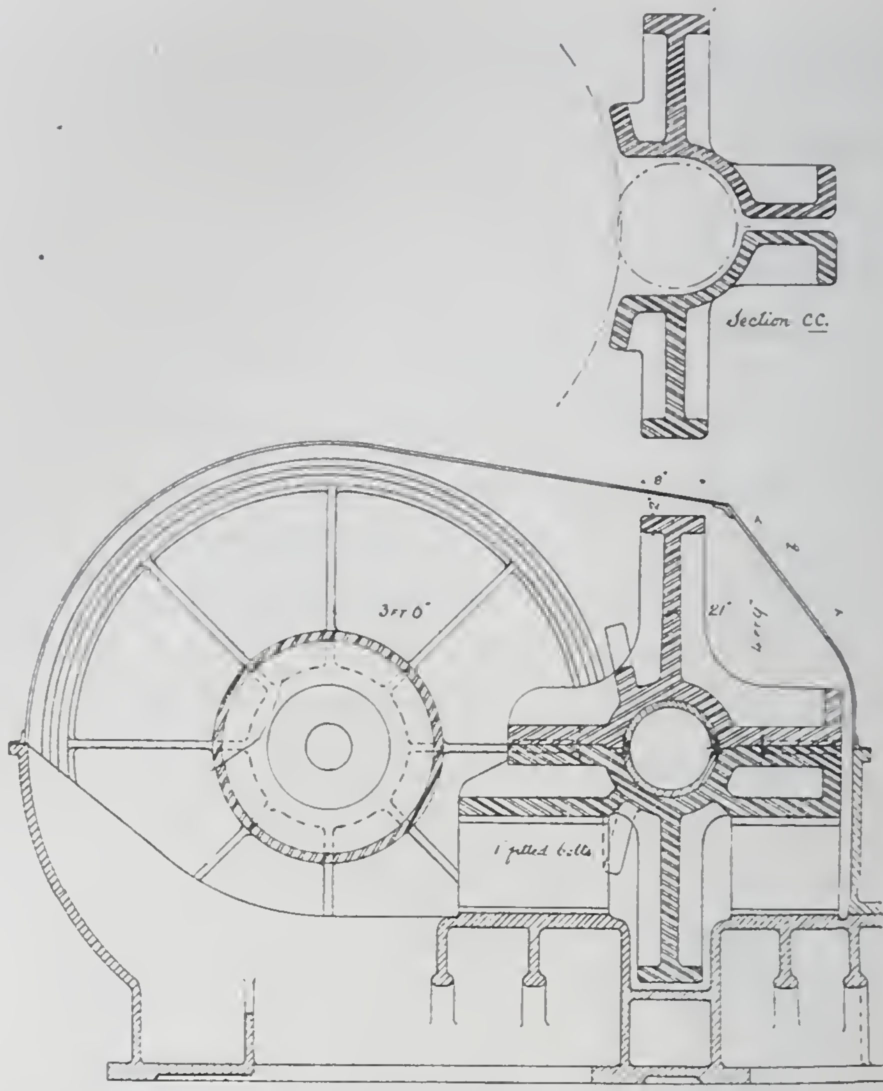

Fig. 2B.

was sent to Mr. Westinghouse and others but not published. While making the investigation the excessive delicacy of such a gear to errors of alignment became very prominent and in this document occurs the first reference to the floating frame—"We have thought very seriously to obviate this difficulty of wear [of the bearings] by carrying this pinion on a very stiff frame so mounted that it would be free to have slight angular motion, both horizontally and vertically, about the center of length of the pinion. The pinion then having three degrees of freedom, would be in proper adjustment however the bearings wore." It was found later that the floating frame would be unstable unless it were deprived of one of these degrees of freedom, but theory also showed that the movement thus restricted was sufficient for the solution of the problem.

The first idea of the floating frame thus suggested itself toward the end of 1905 but did not fully develop for a considerable time thereafter. Fig. 2A and 2B show the first drawing of it, dated September 26, 1906.

The pinion is shown dotted in the front elevation and separately above this elevation. The right- and left-hand helical teeth, with a helical angle of 30 degrees, are merely indicated at B and B′ and it has three journals G, C, G′. The three bearings are seen to be carried in a stiff frame. The front and end elevations show the flexible I-beam supports of this frame. This frame is what we called the floating frame. It does not touch the housing and consequently is free to yield, about a horizontal transverse axis, if there is any change in the resultant tooth pressures. The pinion was also left free to slide longitudinally in its bearings. This drawing shows a solid pinion driven by a square at the end, A. As actually constructed the pinion was made hollow and driven through a flexible shaft and coupling in such a way that no constraint was put on the angular position assumed by the floating frame nor on the free endwise play of the pinion. This construction will be shown immediately.

The diagram of forces is shown in the upper pinion drawing. B E and B′ E′ are the resultants of the tooth pressure, and are normal to the teeth. As the pinion is free to move endwise the longitudinal components, B D, B′ D′, must be equal; and, the parallelograms being similar, it follows that the transverse forces B F, B′ F′, must be equal.

This equality of the forces, it should be noted, is due entirely to the pinion having right- and left-hand helixes and free end play. It is in no way due to the method of mounting the pinion. Thus the equality occurs whether the bearings are rigid or not. But with rigid bearings, if there is misalignment, Fig. 3, the pinion axis sloping up, say, to the left, the centers of pressure, B, B′, will move to the left and the distribution of the tooth pressure will be very uneven. As the total pressures, B E, B′ E′, are not altered, the maximum intensity will be greatly increased. For instance, if the error of alignment is such that the teeth are bearing for only half the length of each helix, the pressure, as indicated by V V, rising from zero at the center to a maximum at the left end ,that maximum will be nearly quadrupled, since the triangular area cut off by V V must equal the rectangular area cut off by U U. A little greater error will give nearly point contact and a tremendous increase of intensity which frequently causes

abrasion and break-down of the rigid gear. How exceedingly slight the error of alignment has to be to produce bad contact I will show shortly.*

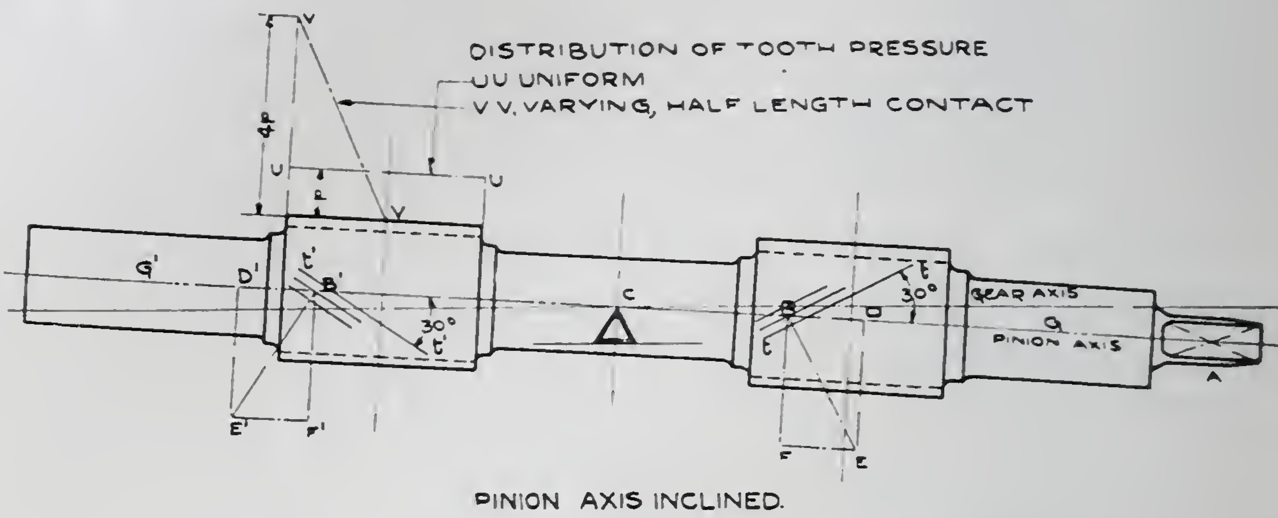

Fig. 3.

But the case is entirely different when the pinion is mounted in a floating frame. Since the floating frame is free to tip about the center C, the moments of the tooth pressures about C must be equal; that is, $BF \times CB = B'F' \times CB'$. But as $BF = B'F'$ we have also $CB = CB'$. I will state four essential conditions which must be fulfilled to insure uniform loading all along the teeth:

1. The equality of the moments of the tooth pressures about C. The preceding paragraphs show that this is automatically fulfilled in a floating-frame gear, but may be entirely upset in a rigid gear.

2. The teeth must be cut with almost absolute perfection and the helix angles of the meshing teeth on the gear and pinion must be equal. (It would be more nearly correct to say that the sum of the right- and left-hand helix angles of the gear must equal the sum of the right- and left-hand helix angles of the pinion. If, say, the helix angles of the gear were each 30 degrees and those of the pinion 29.5 degrees and 30.5 degrees respectively the floating frame would tip one-half degree and correct the error and the equalities $BF = B'F'$ and $CB = CB'$ would not be seriously upset. Of course the intention always is to make equal

*If the pinion is driving two gear-wheels, as in the De Laval gear, there is redundancy of control and it is impossible to predict what the effect of errors of alignment will be.

helix angles, but in one case it was discovered that when the gear was under torque—the teeth bearing well all along—the pinion axis was out of line with the large gear by about two minutes of arc. This was at once traced to a slight error of construction of the machine on which the pinion had been cut. In this case, if the helix angles of the gear were each 30 degrees, the sum of the hélix angles of the pinion was 60 degrees but they each differed by about two minutes from 30 degrees.)

3. The proportion of the length of helix to diameter of pinion must be such that the cross-bending and torsional errors will not be important.

4. If there are three bearings carrying the pinion they must be in line—that is, the floating frame must be very stiff and the pinion properly bedded.

It has frequently been stated that we devised the floating frame to compensate for bad tooth cutting and that it was much simpler to cut the teeth properly and omit the frame. A terrible picture has even been conjured up of the destructive forces which would be induced by the heavy mass of the floating frame swinging about. Nothing will compensate for bad tooth cutting and such an idea never entered our heads. With properly cut teeth the floating frame assumes its correct position and keeps it unmoved.

It will be noted that I have not stated as an essential condition that the pinion and gear must be in line. This condition is absolutely essential with a rigid gear, it requires great delicacy and care to bed the shaft so as to attain it and it is so readily disturbed that it cannot be maintained with any degree of certainty. Hence the tooth pressures of a rigid gear are sure to be highly localized at some time and its safe load is thereby greatly lowered. In the experimental floating-frame gear, tested to 6000 horsepower, the pinion and gear were never in line for I had it set so that the center lines diverged 1/50 inch in five feet nine inches—an enormous error had the gear been rigid. The object had been to design a gear which would run perfectly even if accidentally put out of line and if it would not do this it was no good. To make this clear I show (Fig. 4) a section of the portion of the gear and pinion teeth in contact. The "line of action" on which

the contacts lie is the straight line A B, as the teeth are involute.

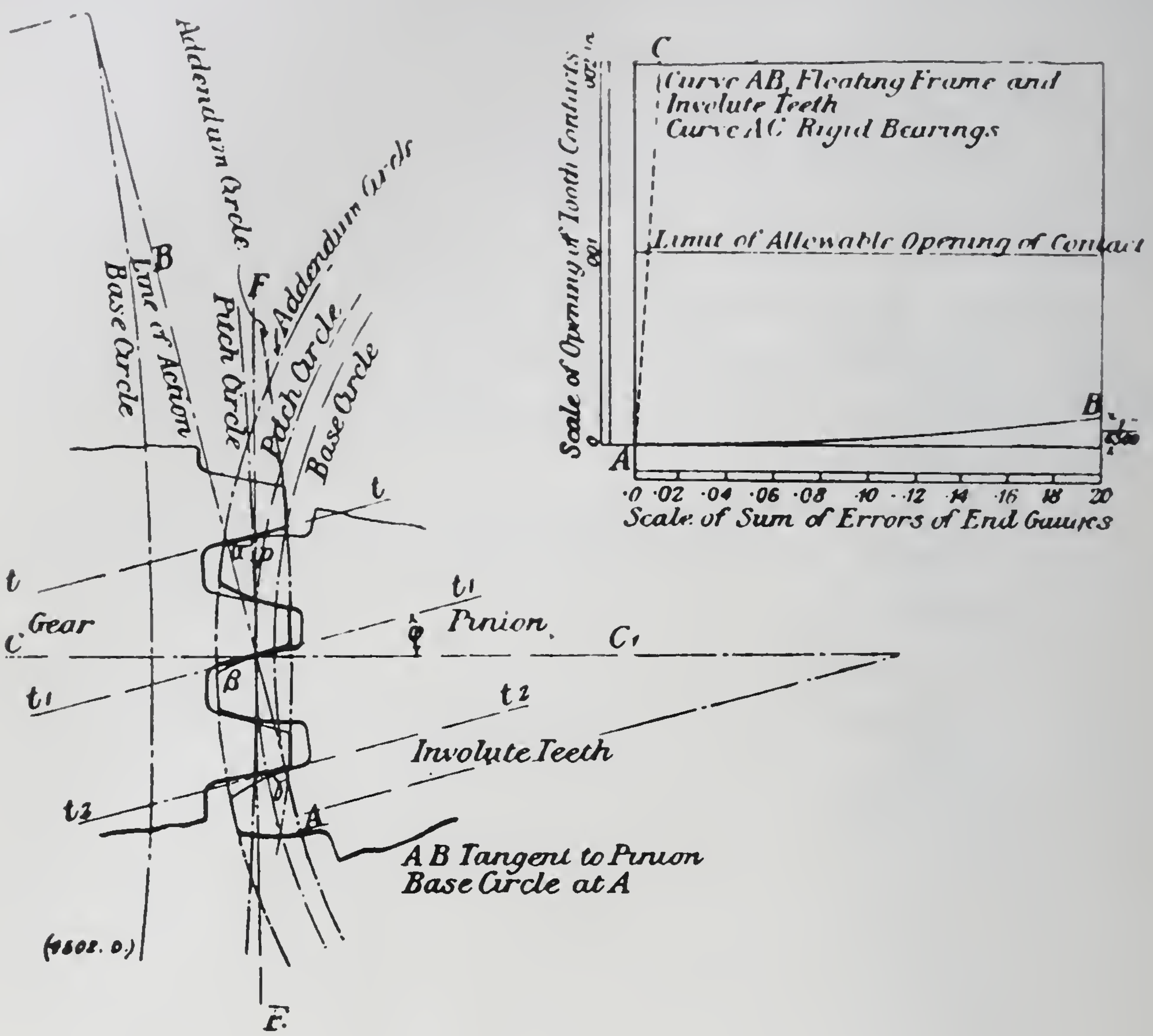

Fig. 4.

Suppose the gear and pinion axes to be parallel and to lie in the same horizontal plane. If now the pinion axis be deflected in a vertical plane, by flexure of the I-beam webs, the tooth pressures would at once bring it back to the horizontal position.

But now let the pinion axis, by warping of the housing, unequal side wear of the bearings, or other cause, be slightly rotated in the horizontal plane. As the pinion has free end play it will adjust itself by a slight screw motion so that the teeth of both helixes are in action—otherwise the longitudinal components of the tooth pressures would not balance and the pinion would not be in equilibrium, but the teeth on one helix would bear hard at the outer end, that distant from the I-beams, and the teeth of the other helix would bear hard at the inner end, near the I-beams. Thus while the parallelograms of forces would remain equal and

of the same magnitude as before, the resultants would not be at the same distance from the I-beams, equilibrium would be destroyed, and the floating frame would tip slightly to bring back the symmetrical position of the resultant tooth pressures. Hence neither a vertical nor a horizontal rotation of the floating frame can exist alone.

It is quite obvious from the diagram of the teeth in contact that there is only one plane in which a rotation of the pinion axis is possible—that is, it must rotate about an axis parallel to the line of contacts, A B. It is well known from the geometry of the involute of a circle that the tangents to the teeth at the points of contact, α, β, γ, are all parallel and normal to A B. They are shown by t t, t_1t_1, t_2t_2. There is no obstruction to sliding of the contact points along these lines in either direction, and a very small displacement will cause no opening of contact.

But if we rotate the pinion axis about a line parallel to A B passing through the I-beams the linear movement of the outer ends of the helixes will evidently be greater than that of the inner ends. Thus, if we continued such a movement but guided it so as to keep the teeth at the inner ends of the helixes in contact, the outer portion of these teeth—since they move in larger curves—will open slightly. It is a simple though somewhat tedious calculation to find how great this opening will be for a given rotation of the pinion axis. I will show the result only. The calculation was made for the experimental gear with an angle of obliquity of tooth action of 14½ degrees.

First let me explain that in the design of the experimental gear provision was made to fit horizontal gages outside the end bearings of the pinion and gear, to measure, even when running, any error of alignment. The gage holes were 9 feet 4½ inches apart. Although these were never fitted, the opening of contact was laid down on an abscissa giving the sum of errors of these gages, the shaft centers being supposed to be wide at one end and close by the same amount at the other.

The opening of contact at the outer ends is shown by the curved line A B (Fig. 4, at the right). A B is tangent to the base line at A, for we have seen that a very small angular displacement produces no opening of contact. When the sum of gage errors

has reached one-fifth inch—an error which will never be remotely approached—the opening of contact is 1/6500 inch. As the opening of contact is proportional to the square of the gage error, at 1/10 inch gage error there is 1/26 000 inch opening and at 1/50 inch error—still very large—the opening of contact has decreased to 1/650 000 inch. All these quantities are far within the limits of the finest machining, at least if we exclude watchmaking.

Furthermore, the teeth are elastic and calculation shows that when loaded to 6000 horsepower an opening of contact of 1/1000 inch would readily be closed. Therefore the closing of these openings shown by the diagram will not sensibly disturb the distribution of tooth pressure. That is, a floating-frame gear can run as well when much out of line as when in line and the bugbear of rigid gears entirely disappears.

The line A C shows the result for a rigid gear of the same dimensions supposing the pinion axis displaced by rotation in a vertical plane. This causes the centers of tooth pressure to shift toward the left or right ends of the helixes, as already explained. The delicacy of adjustment required for proper action is seen to be very great.

Having got rid of this great delicacy it seemed reasonable to hope that a floating-frame gear could safely be loaded to two or three times the continuous safe load for a corresponding rigid gear. This has been fully confirmed by experience.

Even assuming a much smaller load than that to which we tested the experimental gear, it was a safe prediction that we could have replaced the present engines of the *Mauretania* by geared turbines in half the space and with half the weight and giving higher efficiency. Thus a smaller, cheaper, and more economical ship could be built to give the same duty.

The direct-connected marine turbine becomes very uneconomical at reduced speed. The consumption curve (Fig. 5) plotted on r. p. m. is somewhat like A B. The direct-connected turbine being underspeeded to suit the propeller, has the full-speed consumption of steam per horsepower hour, D, and as the speed is reduced this quickly rises to such a point as E. The reciprocating engine usually becomes more economical as the speed is reduced; hence if coal is short much can be saved and

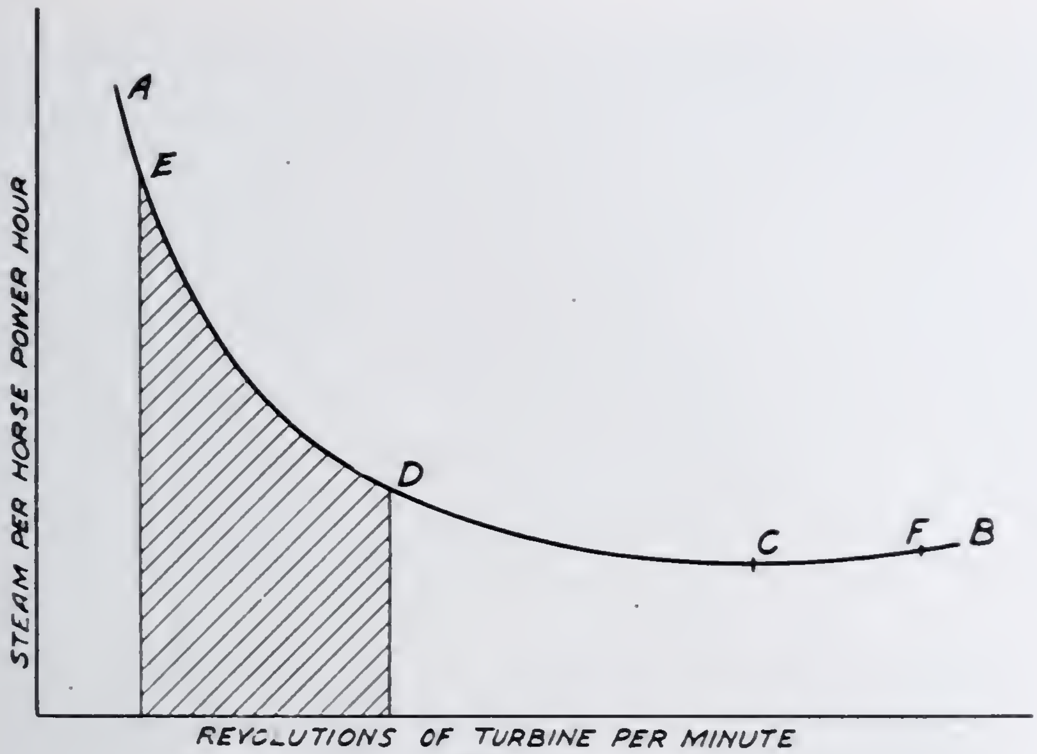

Fig. 5.

port reached more easily by reducing the speed of the ship. But with the direct-connected turbine this important advantage is largely lost. With the geared marine turbine we hoped to make the full power at F, beyond the most economical speed, and the ordinary running speed at C, the point of greatest economy. We hoped that the curve would become so flat that a great reduction of speed could be made without greatly increasing the water rate.

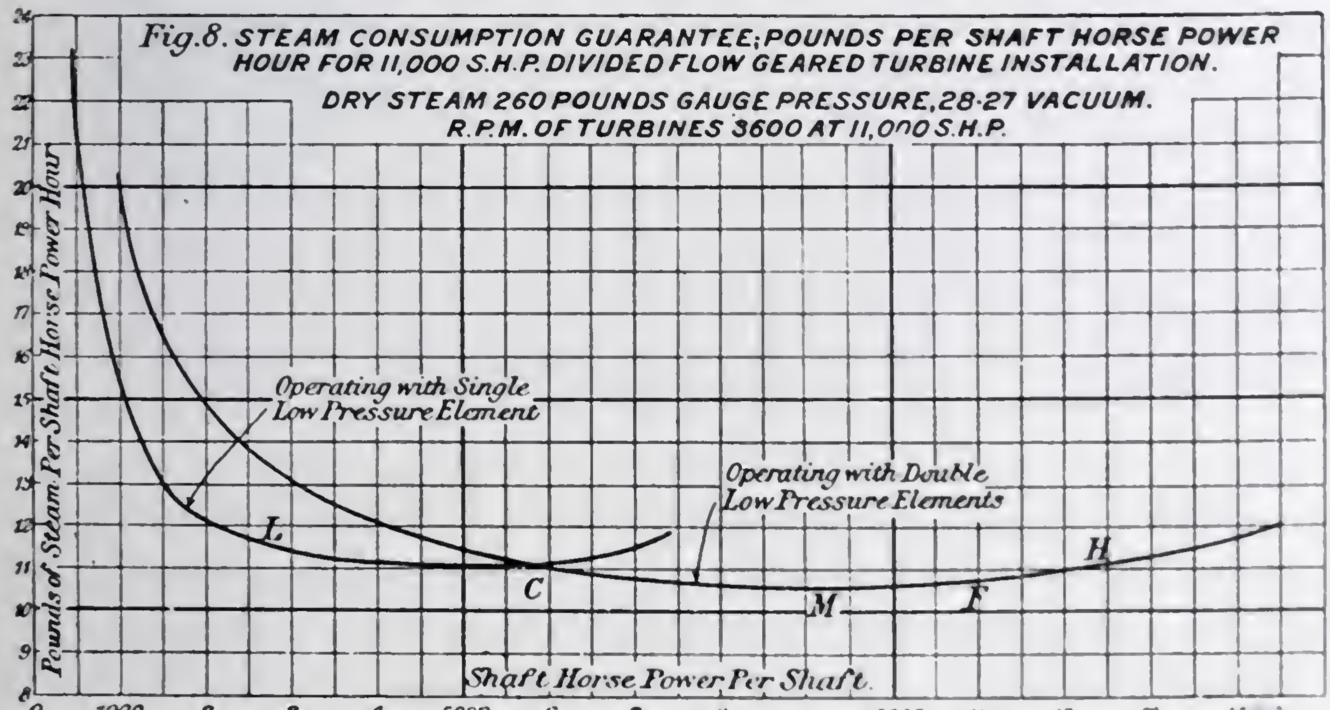

Fig. 6.

Let me show you how completely this has been realized in a recent design of "divided flow" turbine by the engineers of the Westinghouse Machine Company. In Fig. 6, the point for full power is F, falling beyond the point of maximum efficiency M. This and the preceding curve are not directly comparable as this has shaft horsepower as abscissa which greatly flattens the curve. But it is seen that the power can be reduced from 11 000 to about 1200—that is, to less than half speed—without any considerable loss of efficiency, and this good quality of the reciprocating engine is retained. I have not time to enter here on the design of turbine which gives this curve, or its numerous advantages.

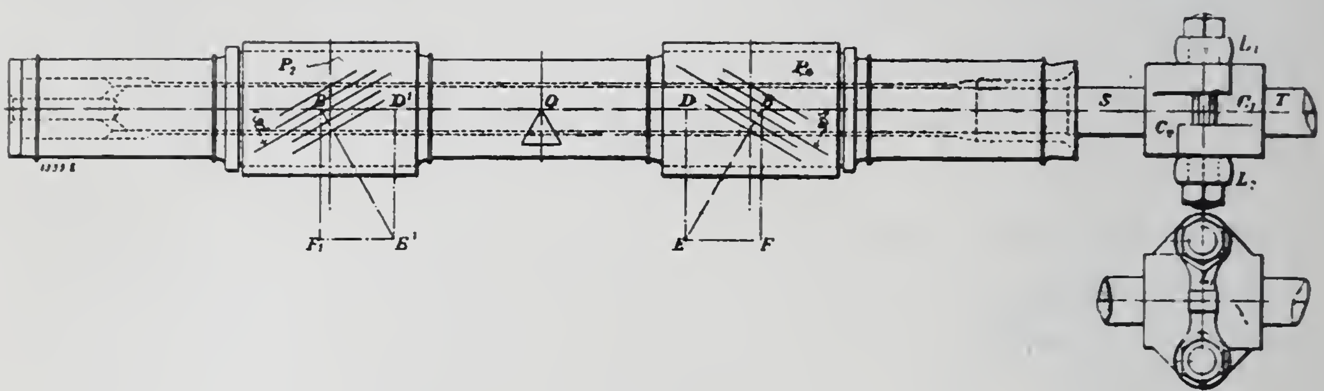

Fig. 7.

Fig. 7 shows the actual method of driving the pinion of the experimental gear. The pinion is bored out and a flexible shaft S passed through it, keyed and bolted at the left end. The turbine shaft T drives this shaft through a coupling. The flange C_1 is mounted on T, and C_2 on S. C_1 drives C_2 through two transverse links L_1, L_2, thus avoiding any sensible longitudinal forces. The floating frame is therefore free to tilt about the center O, and the pinion is free to move longitudinally in its bearings. In subsequent gears the flexible shaft S has been retained but the coupling has been modified, leaving, however, freedom of end motion.

Fig. 8 shows outline drawings of the experimental gear as designed and it will be noted that the floating frame is almost precisely the same as that shown earlier. Transverse horizontal struts, bearing on the housing at one end and on the floating frame at the other, were placed at D, D′, to deprive the floating frame of freedom to rotate about a vertical axis without affecting the freedom in a vertical plane. The floating frame then becomes stable. These struts will be seen in Fig. 9, which shows a per-

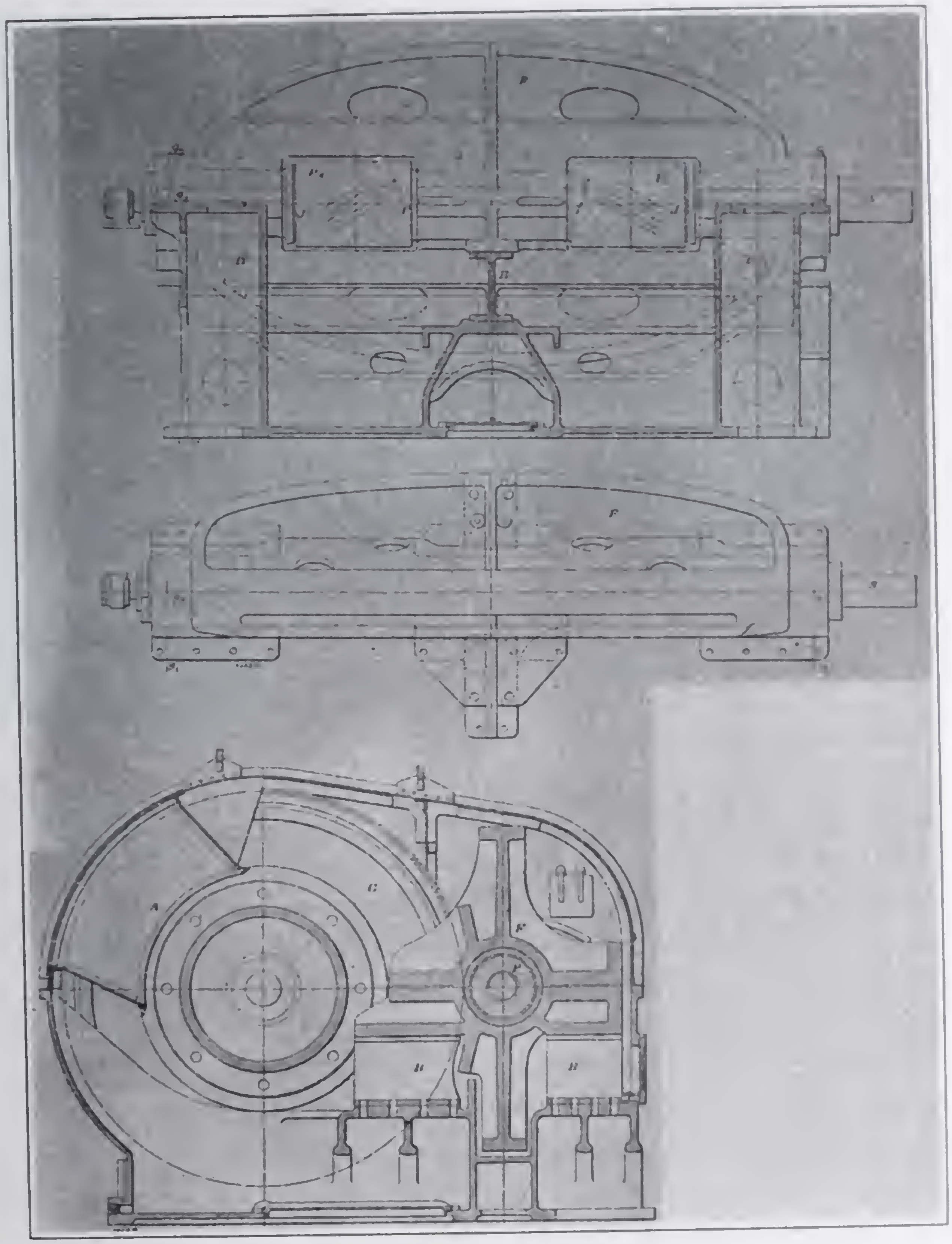

Fig. 8.

spective drawing of the experimental gear with the housing and floating frame partly broken away. The pitch-line diameters of the pinion and gear are 14 and 70 inches respectively. The experiment was on a large scale to make it impressive. The I-beams are plainly seen near the center; one strut is seen near the right.

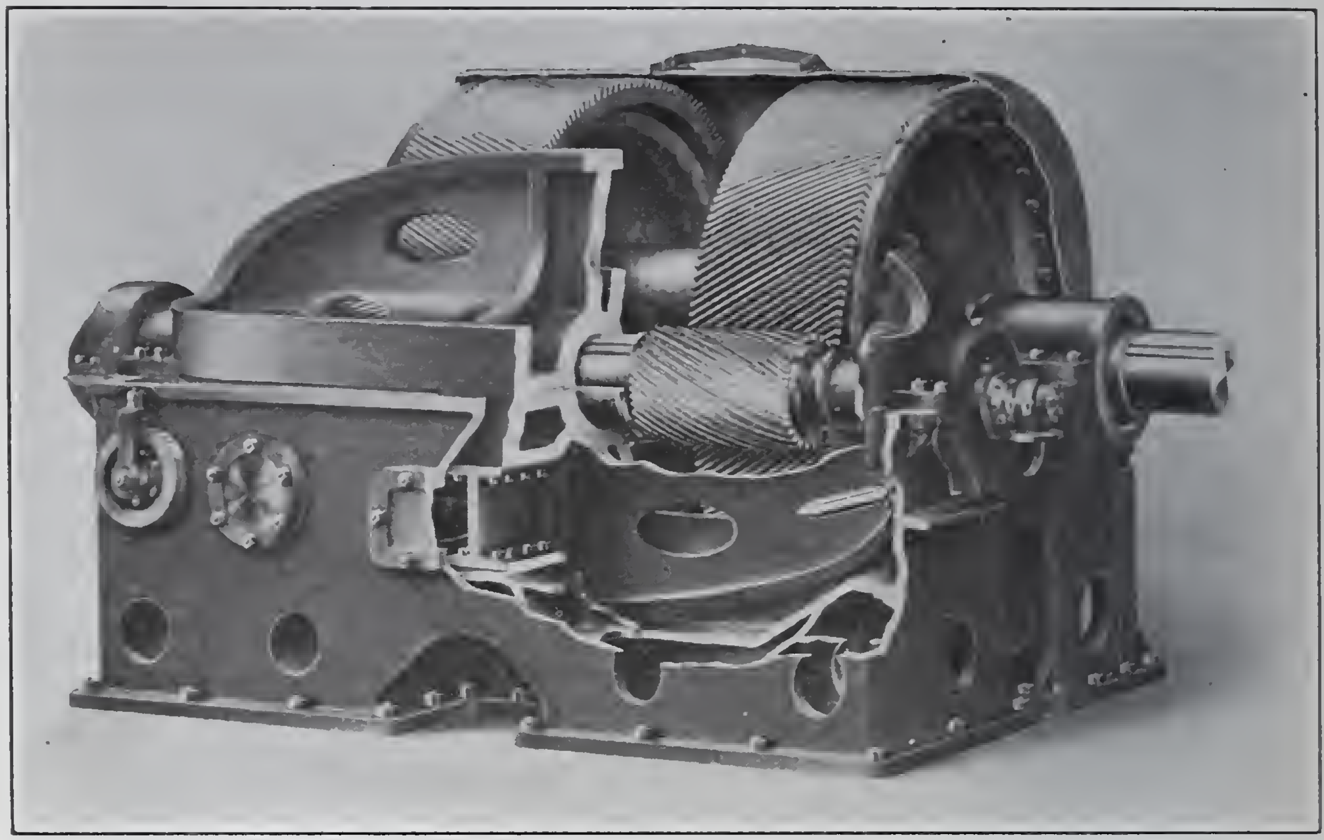

Fig. 9.

and at the left a device for finely adjusting the strut is seen attached to the housing. After adjustment this is locked.

All I have said of the experimental gear, and a good deal more, was published in *Engineering* (London), September 17, 1909, before the gear had been tried. I am pleased to say that all the theoretical predictions made were exactly fulfilled. In fact, experience has shown that the action of the floating frame allows the pinion to be loaded to a much higher horsepower than we dared to dream.

Fig. 10 is from a photograph of this gear which was built by the Westinghouse Machine Company for Mr. Westinghouse; except the gear and pinion. These were made by Messrs. Krupp and the teeth cut by Messrs. Schuchardt & Schuette. For a time all work was stopped due to the panic of 1907. When the air began to clear Mr. Westinghouse called in Mr. Julian Kennedy, a distinguished member of this Society, with whom I went carefully over the design and theory of the device. I am glad of the opportunity to record here that his favorable report helped greatly to get this work started again. I need only quote two sentences— "In a general way, however, I consider this universal mounting of the pinion as a very ingenious idea and one which should give

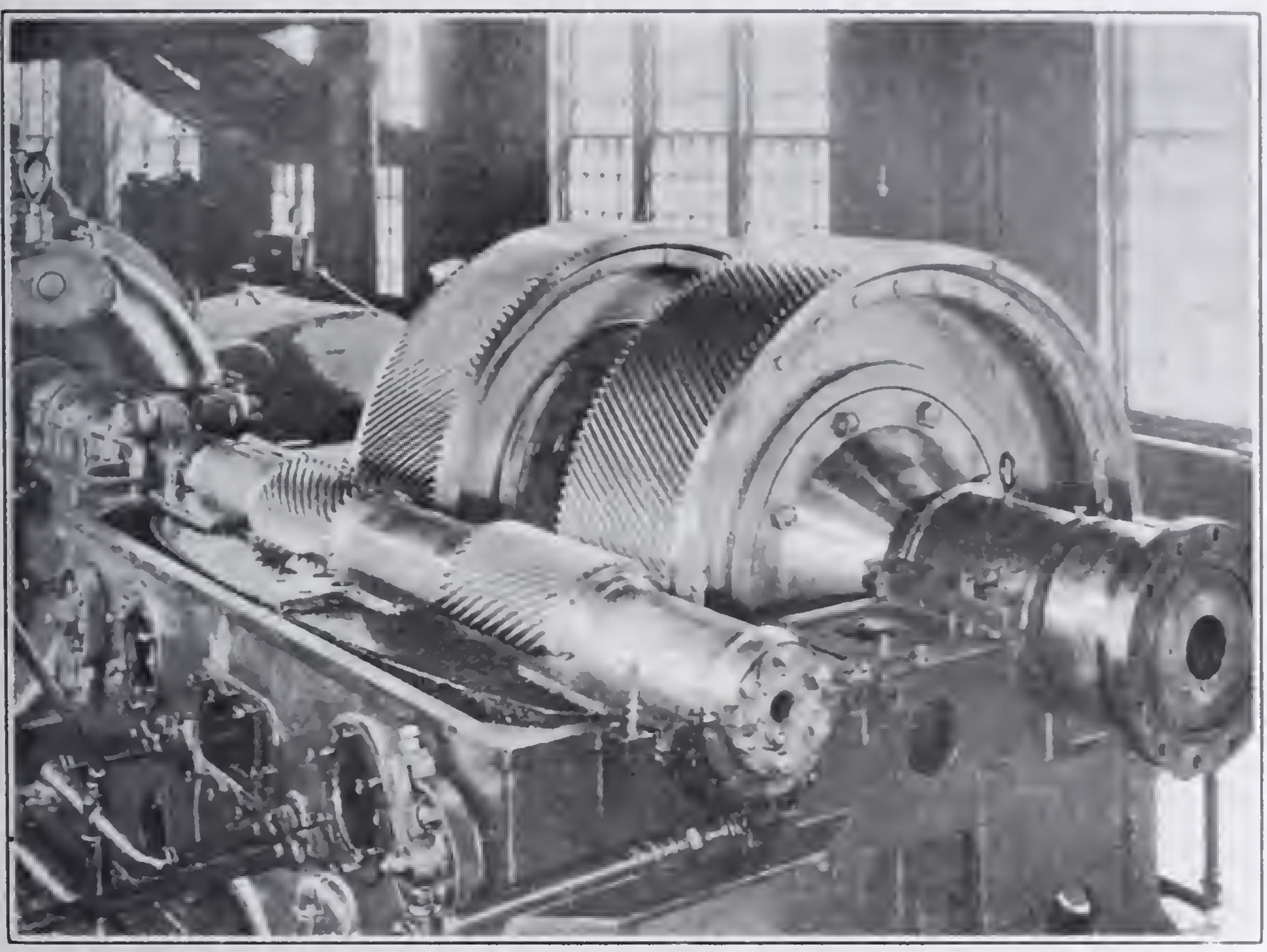

Fig. 10.

satisfactory results in practice". And again—"I think the possibilities of the device are such as to well warrant going to the necessary expense to make a thorough trial of this kind". This greatly strengthened Mr. Westinghouse's hands and he successfully overcame the opposition, proceeded with the construction, and made the tests of the gear, all at a very large cost.

The gear was driven by a 6000-horsepower turbine, actuating the pinion up to 1500 r. p. m. The large gear transmitted the power to a very large water-brake by which it was absorbed and measured. It was started on September 7, 1909, but the principal test was of 40 hours duration, commencing at 3:15 p. m., Saturday October 16, 1909, and ending at 7:15 a. m. the following Monday. It was witnessed by Admiral Griffin, now Engineer-in-Chief, U. S. N., and Commander U. T. Holmes, U. S. N., detailed by the Navy Department for the purpose. The conditions all through were kept almost uniform, the average being 300.6 r. p. m. of the large gear and 6048 horsepower. The frictional and other losses were very slight, the result showing from 98.5

to 99.0 per cent. efficiency. When we opened the doors in the cover immediately afterward, everything was found in perfect condition and from the appearance of the teeth it was evident that the gear was far below its safe load. But all the steam the boilers of the Westinghouse Machine Company could furnish had been used. We know now that we could with safety have loaded it at least three times as much.

In *Engineering* (London), May 5 to June 16, 1916, I have discussed, much more fully than would be possible here, the theory and results obtained from reduction gears and their marine applications but I would like to repeat a few results, theoretical and practical, in addition to the foregoing discussion of errors of alignment, the annulling of which was the object and is the most valuable function of the floating frame.

The cross-bending of the pinion becomes important if the pinion has only two bearings. Consequently a three-bearing pinion will carry a much heavier load than a two-bearing pinion of the same diameter. There is no practical difficulty in securing alignment of the three bearings.

The torsion of the pinion slightly increases the helical angle of one helix and decreases that of the other. This causes the floating frame to tip slightly and it can be shown that this corrects from one-half to two-thirds of the error. The best proportion is to have the length of each helix about twice the diameter.

In floating-frame gears a helical angle of 30 degrees has always been used. This brings sufficient teeth into contact at one time to secure smooth and quiet running, and gives ample longitudinal forces to determine the position of the pinion. In rigid gears this angle is frequently made 45 degrees, no doubt to make these gears more quiet, but this sacrifices much strength of tooth.

Perhaps the most important constant in connection with gearing is what I have called the "power constant". It is:

Power constant $= C = \dfrac{1000P}{D^3R}$, in which

$P =$ horsepower transmitted
$R =$ r. p. m. of pinion
$D =$ diameter of pinion in inches

The factor 1000 is introduced to bring C to a value usually between unity and 10.

A moment's consideration will show that the form of this constant is almost self-evident. Rigid gears frequently give out a ringing noise if driven very fast, but with floating-frame gears no limit has yet been found to the pitch-line speed. These have been run up to about 9600 feet per minute. Hence if in any gear we increase the speed, keeping all the stresses constant, P rises in proportion to R—that is, D being constant, P/R is constant. Again, if we compare two similar gears, one being of twice the linear dimensions of the other, for the same value of R the pitch-line speed will be doubled, which of itself, would double the power. Again, the pitch would be doubled and also the total length of tooth contact, and from the consideration of similar structures it can readily be shown that the corresponding total tooth pressure would be quadrupled. This would again quadruple the power so that the power transmitted by the larger gear is eight times the corresponding power of the smaller, or proportional to D^3. This rule can readily be shown to apply to any ratio of increase. Hence for R constant, P/D^3 is constant and the form of the power constant is established.

In the *Engineering* (London) articles referred to above I gave the two highest power constants, for rigid gears, which I have found. These were both for marine gears. The *Vespasian*, the first geared ship engined by the Turbinia Works Company, gave a power constant of three. The sister ships *Ciudad de Buenos Aires* and *Ciudad de Monte Video* gave 2.58 for the pinion of the high-pressure turbine and 1.29 for that of the low pressure. These were all the highest measured-mile trial results of a few minutes duration. From published results it appears that the gears of the *Vespasian* operate usually with a power constant of about 1.85 and no doubt the sister ships referred to also have much lower power constants when in service.

Power constants of 3.5 and 4 are perfectly safe for floating-frame gears. Indeed, in the twelve-hour full-power trial of the United States collier *Neptune* the average power constant was 5.22. She has been in service for over two years and is frequently used at full power. Between April 1, 1916, and March 31, 1917,

she steamed about 40 000 nautical miles, the speed for most of this distance being between 12 and 13 knots, giving a power constant varying from 3.5 to 4.3.

The earlier gears had lower power constants. As experience was gained this value was raised. The following table shows those with higher power constants which have been running longest, excluding that of the *Neptune*. They are land gears, as for them the service is most continuous.

TABLE I

No.	Started	Running time: Hours per day	Running time: Days per week	Power constant
1	April 4, 1914	16	7	3.54
2	April 12, 1915	24	7	3.24 to 4.05
3	Nov. 1, 1914	12 to 16	6	Up to 4.05
4	June 5, 1914	24 in summer	6	3.41
5	Aug. 14, 1914	24 in summer	6	3.41

Every one of the floating-frame gears put out by the Westinghouse Machine Company is in perfect condition to-day except the gears of the S. S. *Malmanger*—torpedoed before she had quite crossed the Atlantic—and the experimental gear which, having served its purpose fully, has been scrapped; being only a five-to-one reduction and very large it was not found salable. All the rest are running except the first gears of the *Neptune*. These were replaced by others, as it was determined to change the ratio of reduction in order both to raise the speed of the turbine, and to lower the speed of revolution of the propeller to that of her sister ship. The action of the gears had been entirely satisfactory.

The fact that such high power constants can be used is convincing proof that practice agrees with the foregoing theory of the action of the floating frame. Further confirmation should be evidenced by the wear on the teeth, which should be extremely small if pressure distribution is good. This is abundantly proved by the first gear put in operation. It was installed by the Commonwealth Steel Company, Granite City, Ill., and started on March 31, 1911. When examined on April 30, 1916, after over five years service, the *scraper* marks, originally of imperceptible

depth, were still visible on the gear teeth, and the driving face of the pinion teeth had taken on a very high polish. The gear was as quiet as at first. When installed it ran five and latterly six days and nights per week. It may be objected that this early gear had not a very high power constant, but at maximum load this was 1.68. Not many rigid gears exceed this value in regular service.

It has been stated by some rigid-gear builders that bad distribution of tooth pressure is corrected by a shift of the relative position of the journal and bearing centers, thinning the oil film or changing the position of its minimum thickness. This is wrong for two reasons: 1. Departure from the normal conditions in the bearing necessarily indicates an abnormal tooth pressure distribution. Consequently it could not be a means of maintaining uniform distribution. 2. The minimum film thickness will never exceed 1/500 inch and will usually be much less. As I have shown elsewhere, Prof. Osborne Reynolds's theory of lubrication shows that even for large changes of journal pressure the change in the oil film is excessively minute. The hope that this adjustment would be effective is founded on desire, not on knowledge. If it were correct, rigid-gear manufacturers could use much higher power constants than they do.

Increase of the safe power constant obviously means increase of the safe load which can be put on a given size of gear, and unless the change of design by which this increase is effected involves a proportional increase of weight, the weight per horsepower is decreased. Since weight and compactness (especially in marine applications) and safety in all cases, are the principal considerations, it follows that the higher the power constant a gear will safely bear, the better. Therefore gears, whether of the same or different types, should be compared principally by the safe value of the power constant, due allowance being made for any change of weight or bulk inherent in the design. In comparing rigid and floating-frame gears the rise of the safe power constant offsets many times the increased weight and bulk; indeed, for a large ratio of reduction these increments are insignificant. In one marine installation the weight guaranteed by the Westinghouse Machine Company was 26 000 pounds; while two

competing rigid-gear builders offered 40 000 and 50 000 pounds respectively. The floating-frame gear was accepted and built at 1000 pounds less than the guarantee.

In a recent paper read before the Society of Naval Architects and Marine Engineers, New York, by Mr. Robert Haig, till lately Chief Lloyd's Surveyor in Philadelphia, now an officer of the Sun Shipbuilding Company, it is stated that there have been failures with gears and there may be a doubt of their lasting qualities, but that high-class alloy steel is now in use. This statement in no way applies to floating-frame gears, in which high-class alloy steel is *not* used either for the pinion or gear rims. These are always made of a good quality of mild carbon steel. This commercial steel is more readily procurable and much more reliable and does not fail, though carrying more than twice the load of gears of the same dimensions which have no provisions for securing good distribution of tooth pressure and in which, consequently, imperfect contact is sure sooner or later to occur.

The adoption of high-class alloys is an admission of great trouble from errors of alignment of the gear and pinion. It does not remove the cause of failures—that is, error of alignment. It materially increases first cost and the harder material will intensify the localization of pressure, and great danger of abrasion will remain.

The floating frame has been criticized as being a complication but that is not the opinion of the engineers of the Westinghouse Machine Company, who have much experience in building both rigid and floating-frame gears, and know the great care which must be exercised to attain as perfect alignment as possible in the rigid gear. After rough machining, internal stresses must have time to relieve themselves or, better, be relieved by annealing, after which the finishing cut is made. The bedding of the pinion is tedious as only by trial can it be found whether the teeth bear all along. Again, they found that the greatest care must be exercised in bolting down the gear housing—as predicted, the screwing down of a holding-down bolt readily puts the gear and pinion axes out of line. Hence one of the reasons for low power constants and high-class alloy steel of rigid gears.

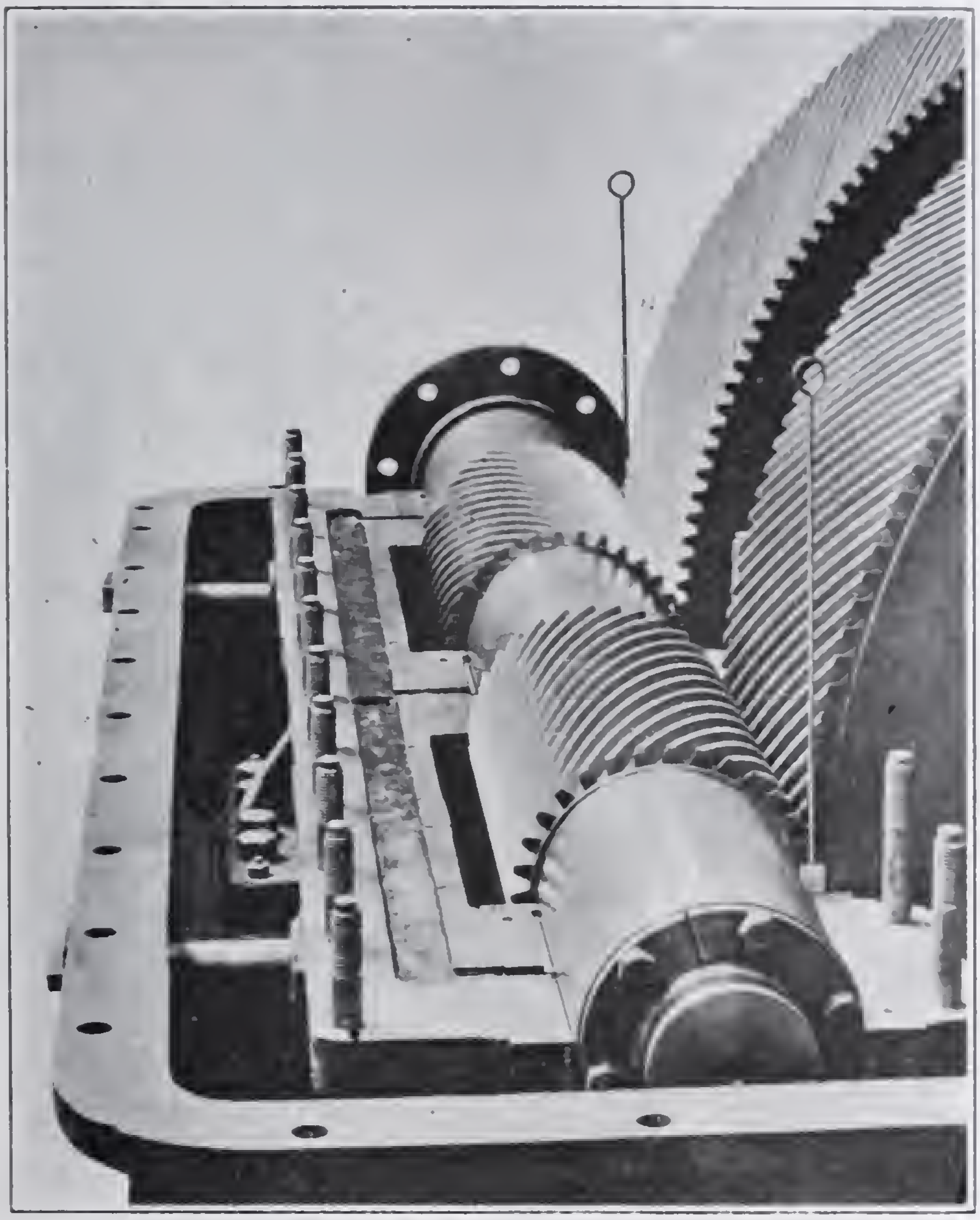

Fig. 11.

In great contrast is the simplicity of lining up the floating frame (Fig. 11). Parts just outside the teeth are very accurately turned to diameter on gear and pinion; after the ordinary bedding of the journals, the floating frame is adjusted by block gages bearing on the turned parts as shown, and bolted down.

The first merchant ship fitted with propelling machinery by the Westinghouse Machine Company was the *Malmanger*, an oil-tank steamer of 12 650 tons displacement, 2900 shaft horsepower, and 10.5 knots. Fig. 12 is a plan of her engine room, showing the cross-compound turbines and also two first reduction gears and one second. The turbines are aft of the gear as this allows the propeller shaft to be drawn inboard without disturbing the

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ERRATUM

Insert in Proceedings of the Engineers' Society of Western Pennsylvania, April, 1918, following page 276

Titles of Fig. 6 and Fig. 7 are transposed.

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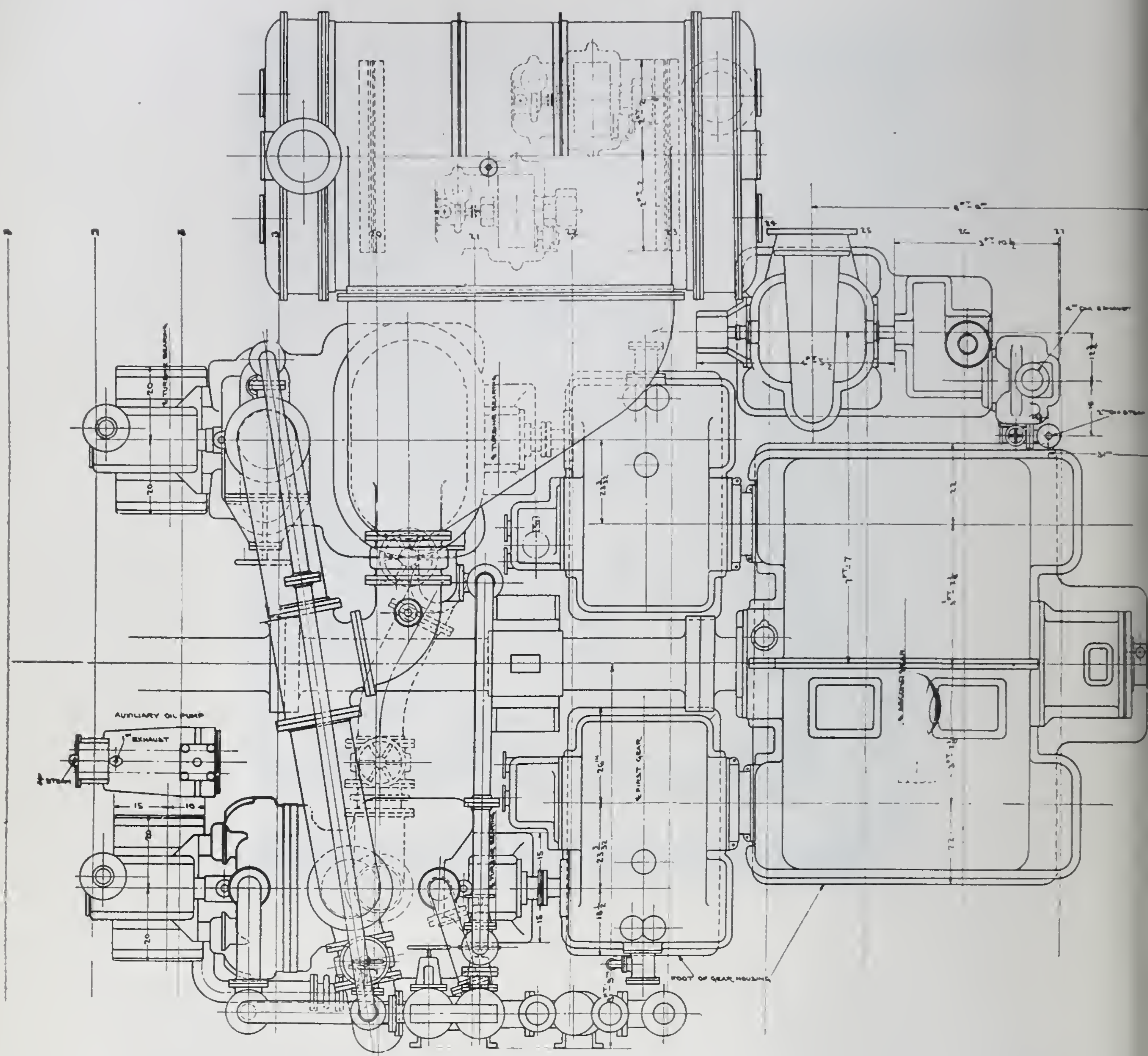


Fig. 12.

gear. The high-pressure turbine has impulse and reaction blading; the low-pressure only reaction blading. Each turbine casing has an astern impulse turbine. The astern power guaranteed is 60 per cent. of the full ahead power with the same flow of steam. For full speed ahead the turbines run at 3860 r. p. m. and the propellers at 75 r. p. m. The *Malmanger* was torpedoed southwest of Ireland on her first voyage across the Atlantic.

On the *Golaa,* a sister ship, reversing from full speed ahead to full speed astern took 55 seconds; full astern to full ahead, 45 seconds.

The first reduction gears of the *Malmanger* are shown partly in section in Fig. 13. The upper view shows the floating frame. The cylinders F constitute a dynamometer for measuring the power transmitted. (This developed from a form of the floating frame devised by Mr. George Westinghouse, in which all the bearings were supported by hydraulic cylinders. A good many

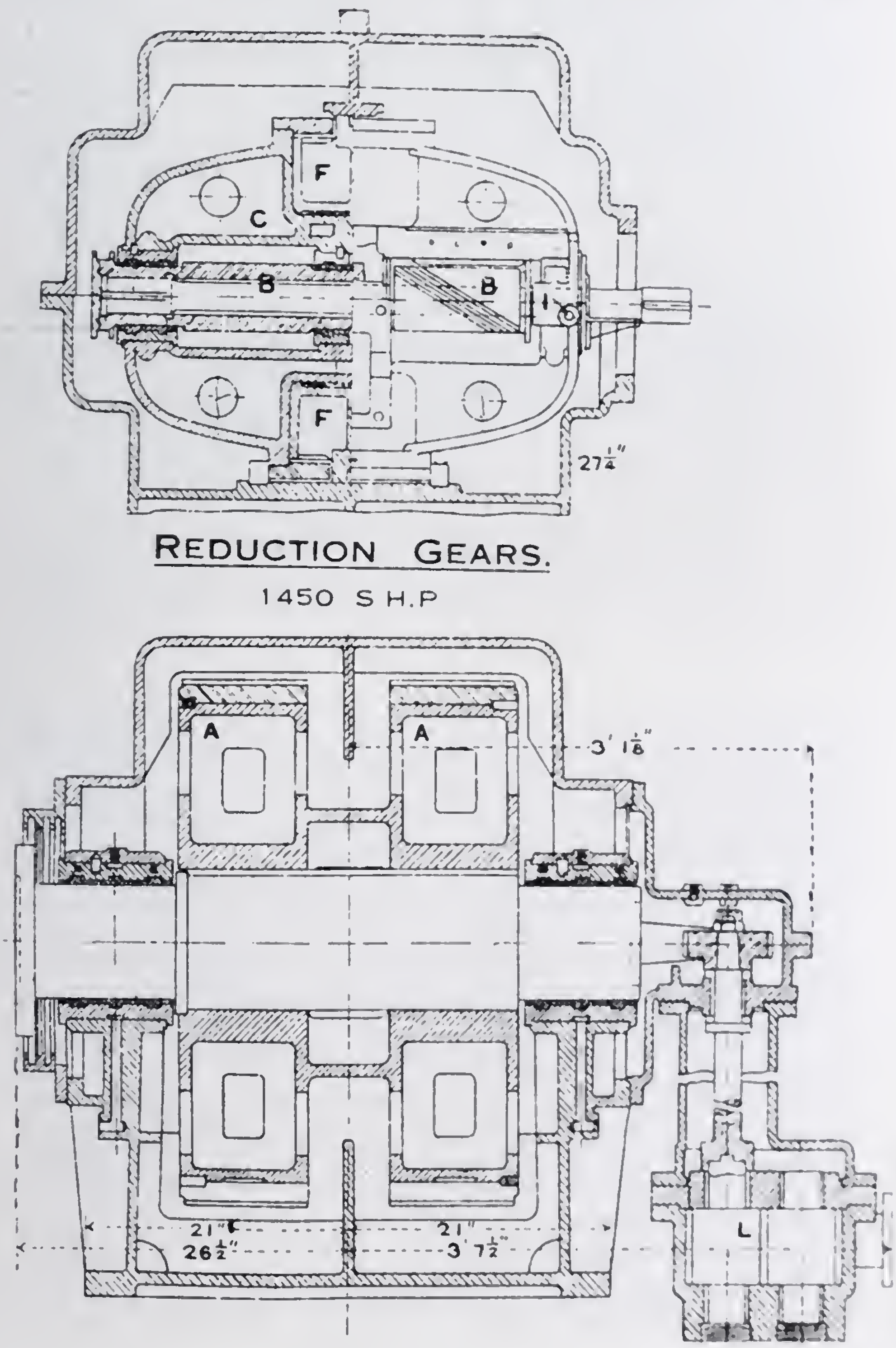


Fig. 13.

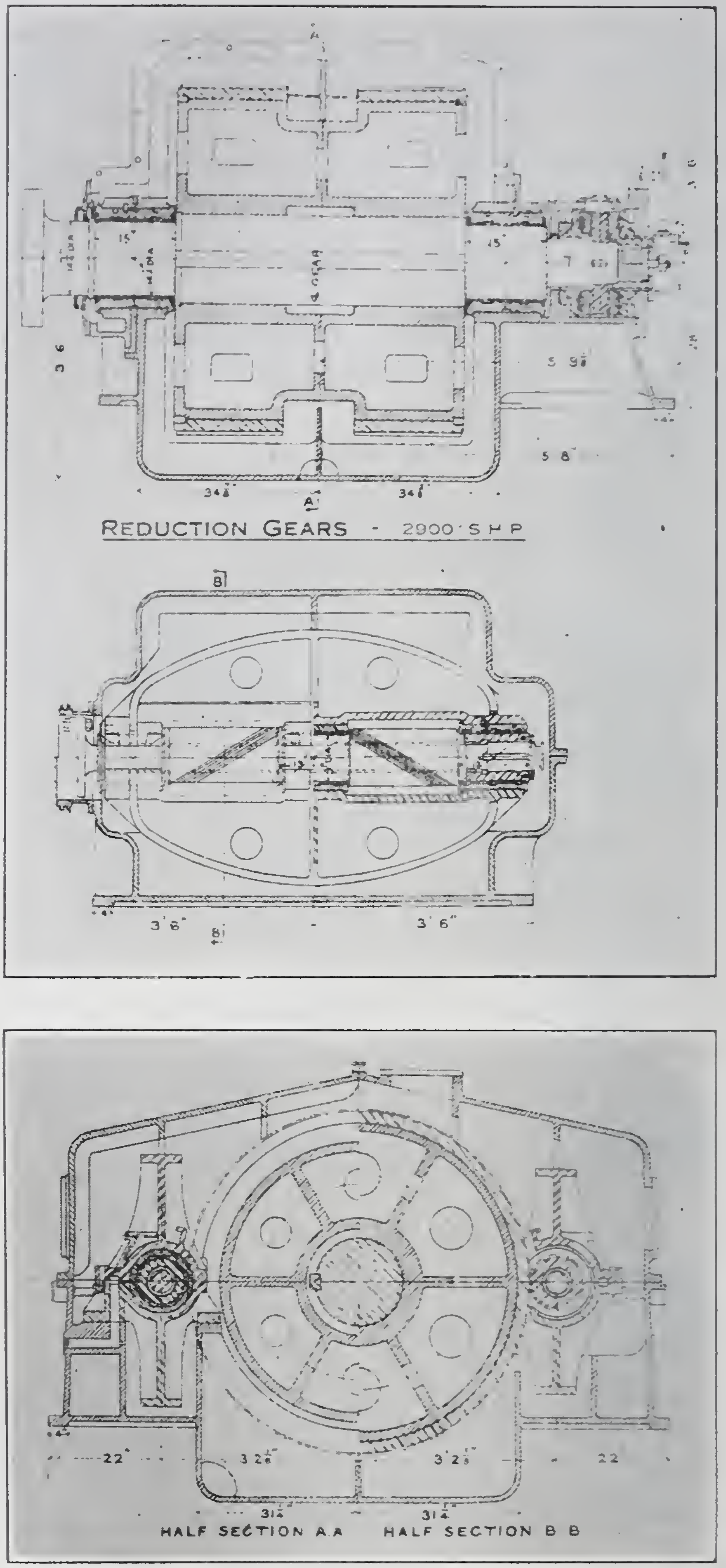


Fig. 14.

of these are in operation, but the earlier and simpler I-beam frame has now been reverted to.) When the power is not being measured the floating frame is bolted to the housing. When it is desired to measure the ahead power these bolts are very slightly turned back so that by pumping oil into the lower cylinder F, the floating frame is oil borne. Obviously, from the oil pressure, diameter of oil cylinder, and speed of pinion, the power can be calculated. The upper cylinder F is similarly used to measure the astern power. The lower view shows a section of the large gear and also the rotary oil pump L, driven by screw gearing from the gear shaft. The flow of oil to all high-speed bearings and to the teeth must be copious. It is projected on the teeth just as they are coming into mesh so that centrifugal force cannot prevent thorough lubrication.

The second reduction gear is shown in section in Fig. 14. The upper view shows the Kingsbury thrust bearing, invented by Mr. Albert Kingsbury, one of the best known members of this Society. This bearing takes the thrust of the propeller which drives the ship, replacing the numerous collars of the ordinary thrust block by a single collar, thus saving much space and weight; and, more important still, it is more definite and therefore safer in its action. The thrust of the collar is taken by plates so mounted that they automatically adjust themselves to maintain an oil film between the collar and plate. This thrust has given splendid results and has been widely adopted. It is a practical application of Prof. Osborne Reynolds's theory of lubrication, already referred to. It may not be uninteresting to note that both in this device and in the floating frame, success is attained by removing unnecessary geometrical restraints, a principle of machine design strongly inculcated by my old teacher, Lord Kelvin.

The lower view shows a transverse section of the second reduction gear. Each first reduction gear drives one of the pinions.

The *Maui* is a twin-screw passenger and freight ship of 17 500 tons displacement and 17.5 knots speed, running between San Francisco and Honolulu. It was built at the Union Iron

Works for the Matson Navigation Company. The turbines are cross-compound and reversible. To conform to the limitations of the engine room, the high-pressure turbine actuates a pinion almost on top of the gear (Fig. 15). The low-pressure pinion is

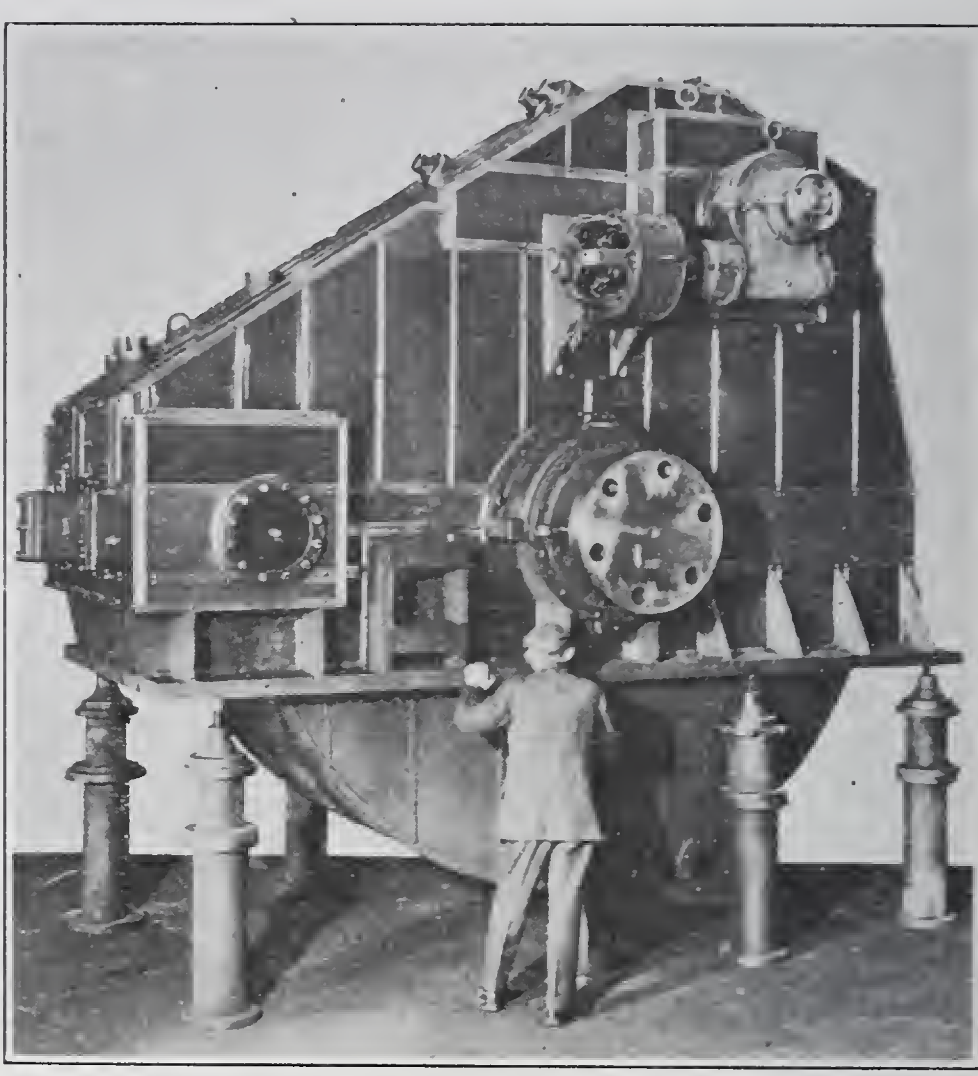


Fig. 15.

level with the gear shaft. The view also shows the motor actuating the high-pressure pinion for turning the gear, turbine, and propeller, when overhauling. In Fig. 16 the gear cover is removed showing the low-pressure pinion and floating frame. The Kingsbury thrust block, closed, is also shown. The maximum guaranteed continuous horsepower is 12 500 with 2120 r. p. m. of turbine and 133 r. p. m. of propeller; but usually less power (8500 horsepower at 120 r. p. m. of propeller) is used. She has proved remarkably economical. A single-screw sister ship, the *Matsonia,* is fitted with reciprocating engines. The owners, by careful observation of five round trips of each, find that the geared-turbine-driven *Maui* is 22.87 per cent. more economical.

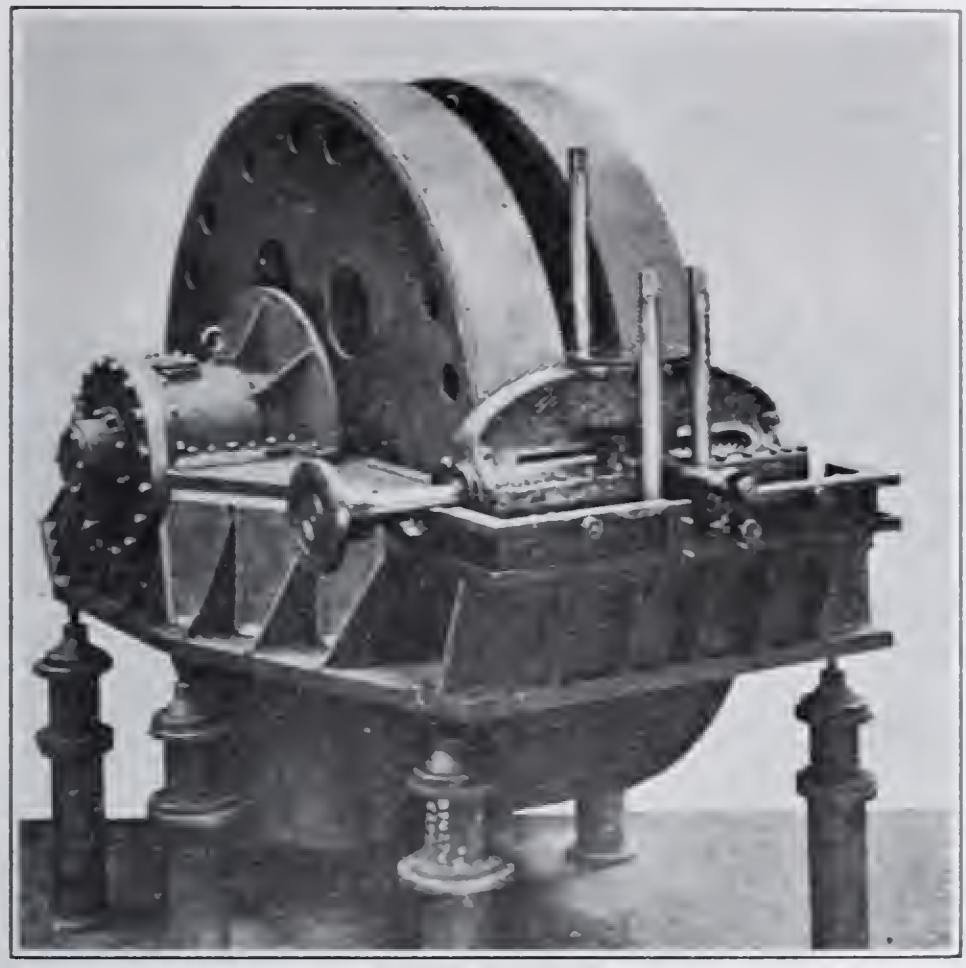


Fig. 16.

This is in spite of the fact that the oil burners of the boilers of the *Maui* are defective, and the propeller pitch is too coarse, holding down the propeller speed to 106 r. p. m. instead of 120, which would ordinarily have been used. When these parts—neither of which came within the Westinghouse Machine Company's contract—are made satisfactory no doubt a further economy will be effected.

The U. S. battleships *Pennsylvania*, *Arizona*, and *Mississippi*, have two sets each of cruising gears actuated by cruising turbines. A clutch between these gears and the main turbines is thrown in when the ship is cruising at low speed. The steam is first expanded through the cruising turbines before it enters the main turbines, and thereby a great increase of economy is effected when cruising, and the radius of action of the ship is greatly increased.

The U. S. S. *Melville*, named after Rear Admiral George W. Melville, and used as a mother ship for destroyers, is a single-

screw, geared turbine ship of 4000 horsepower, the high- and low-pressure turbines each actuating a pinion, and these in turn driving one large gear on the propeller shaft. The *Melville* is now in European waters helping to fight the submarine.

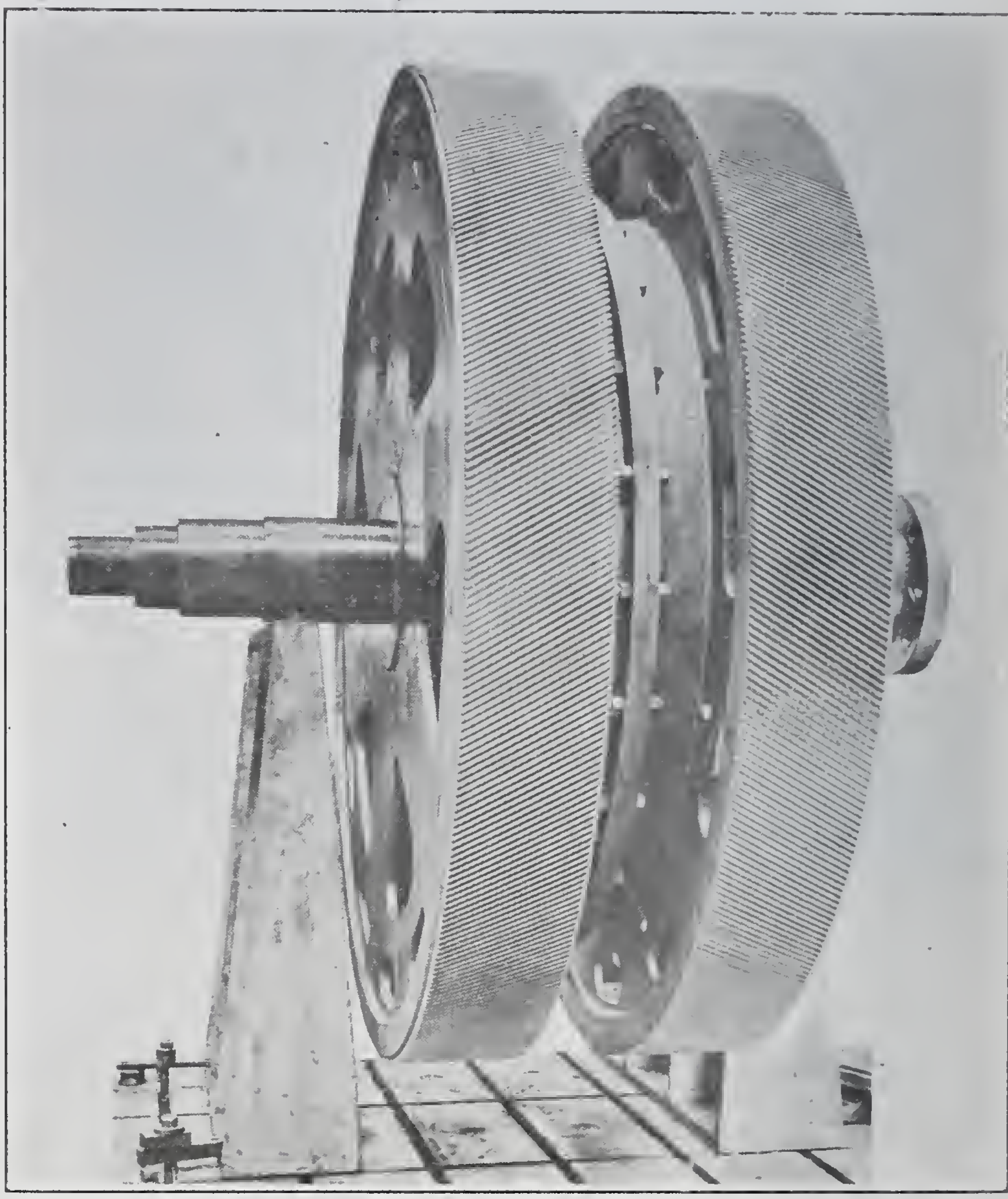


Fig. 17.

The gear shown in Fig. 17 is one of two to be installed in a Swedish cruiser of 22 000 horsepower. Each will transmit 11 000 horsepower from the pinions of the high-pressure and low-pressure turbines. Its pitch diameter is 10 feet 9.326 inches and it has on each helix 448 teeth of 0.907 inch circular pitch. The breadth of each helix is 15½ inches.

In Fig. 18 and 19 are shown two views of a double-reduction marine gear, with two first-reduction gears complete, and with two second-reduction double-helical pinions driving one double-helical large gear on the propeller shaft. In Fig. 18 the upper halves of the two floating frames on the near side of the second-

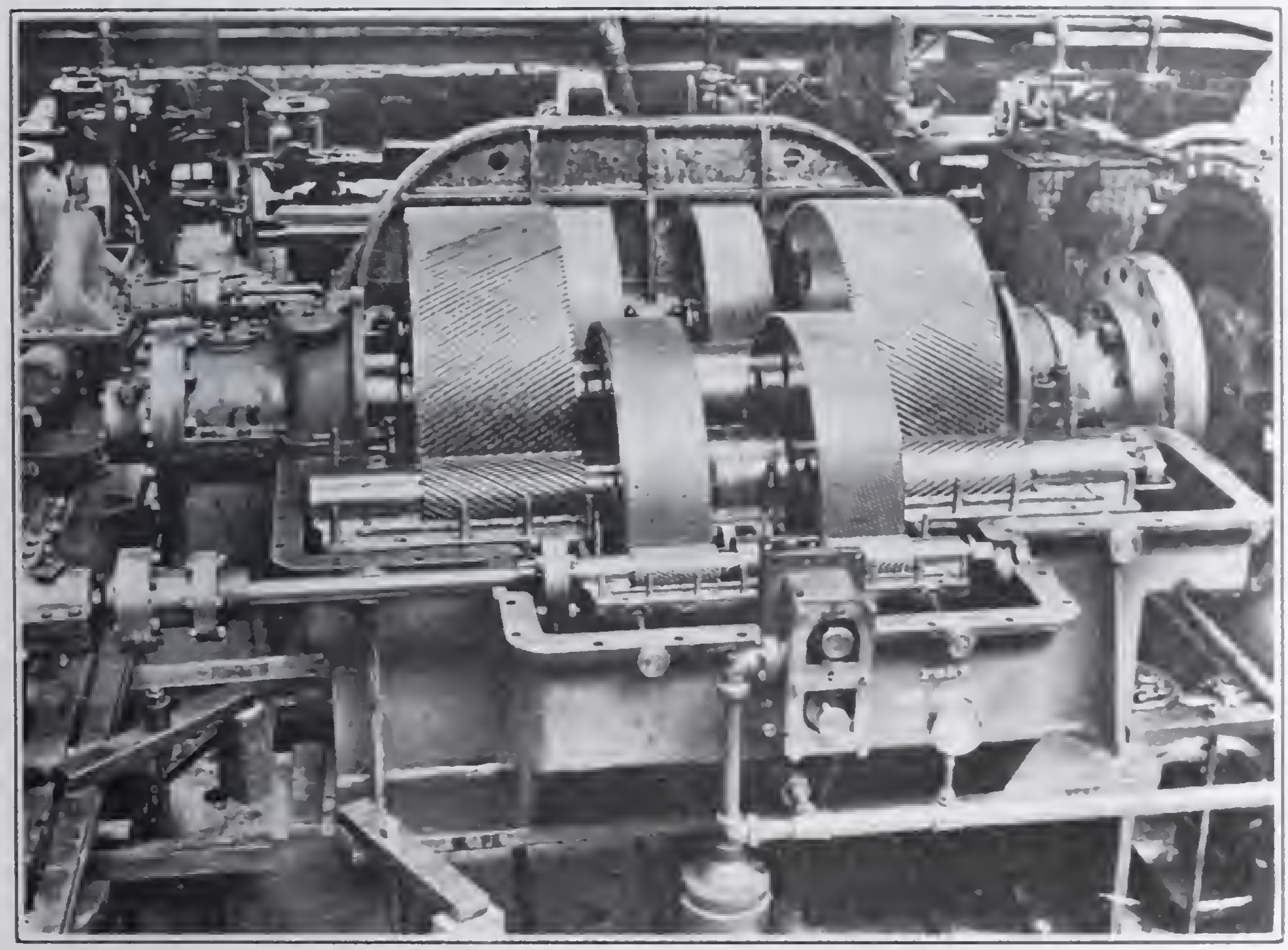


Fig. 18.

reduction large gear, are removed, but the upper half of the second-reduction floating frame in the far side is in place. It will be seen that this floating frame carries the large gear of the first reduction and the pinion of the second reduction. The whole gear is in one housing and forms a light and compact design. The gear is of 3400 horsepower at 3740 r. p. m. of the small pinions. The reduction is to 72 r. p. m. of the propeller shaft, or a total ratio of reduction of nearly 52 to 1. In Fig. 19 is a view looking down on the same gear, the upper halves of each of the four floating frames being removed.

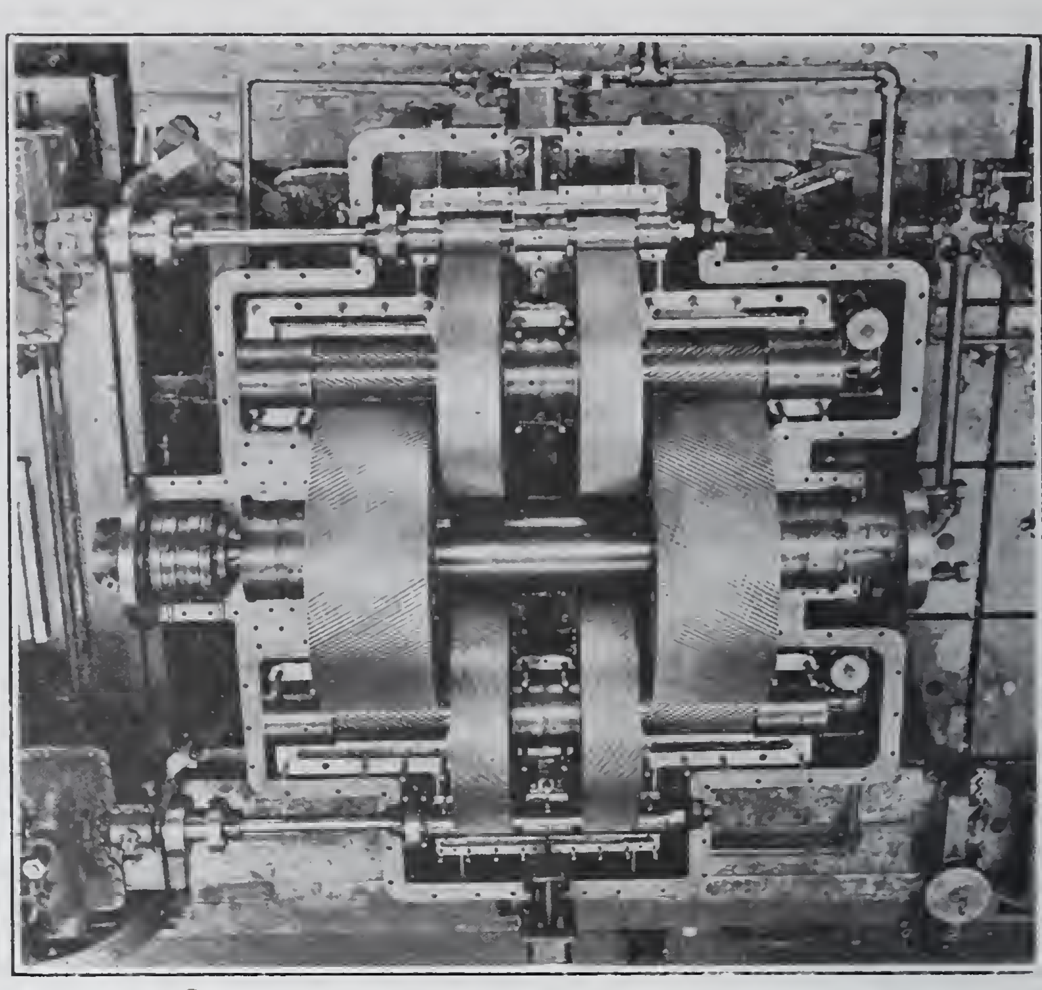


Fig. 19.

Fig. 20A and 20B show a double-reduction gear, arranged vertically for a merchant ship.

As is usual with most things new, the merits of the floating frame came only gradually to be understood and progress at first was slow. This accelerated later, so that for some years orders have been placed more rapidly than the facilities for turning them out could be expanded. Even before the recent large orders were placed by the United States Government, the gears undelivered totaled about 1 000 000 horsepower. Nearly all the early gears were for land installations, though the primary problem we had in view was the improvement of marine propulsion. Gradually the orders for marine applications grew, and latterly far exceeded those for land. In fact, the marine field so expanded, not only for gears but for turbines, condensers, pumps and other auxiliaries, that for a long time much land work which offered has been refused by the Westinghouse Machine Company, and not nearly all the marine work could be taken.

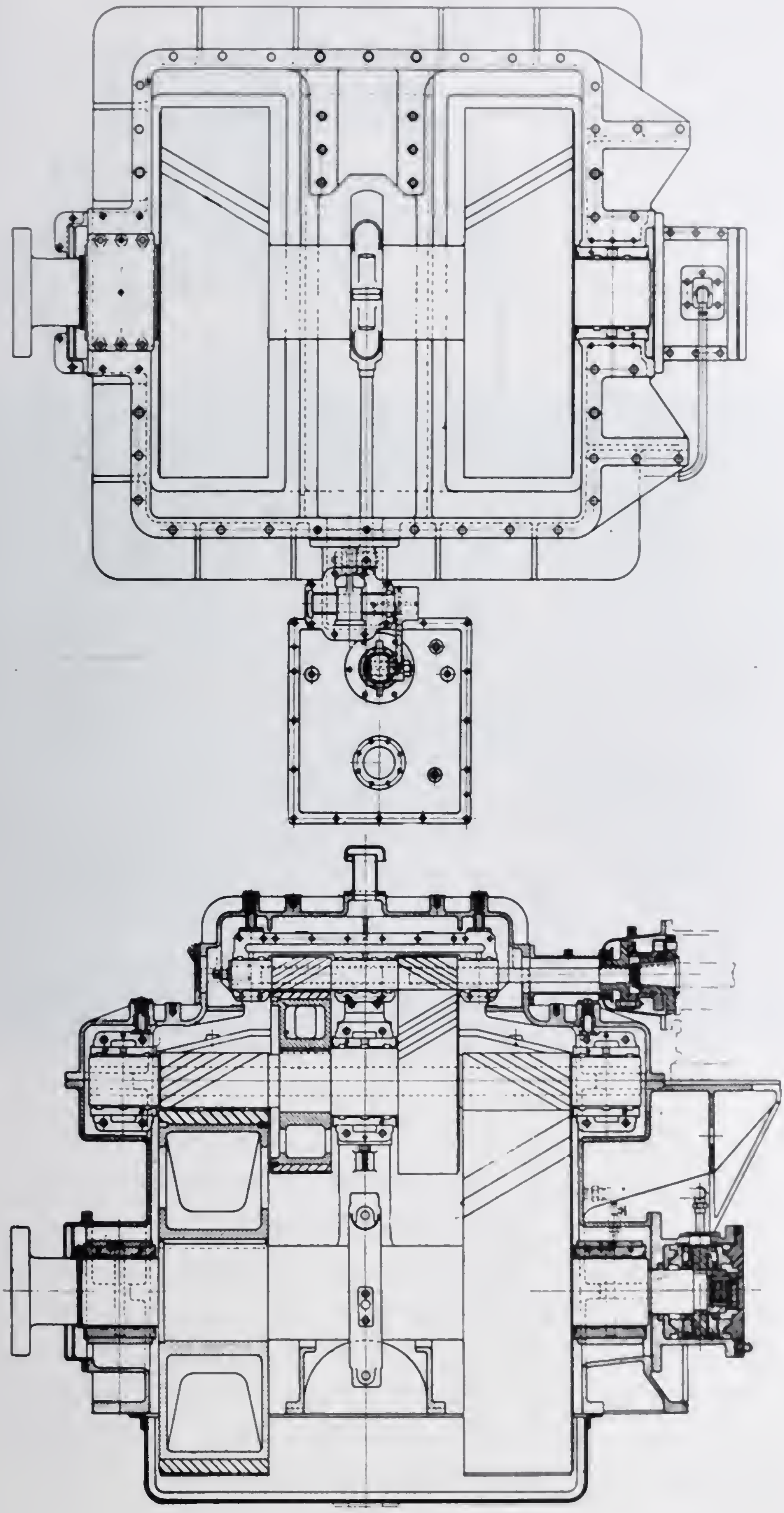


Fig. 20A.

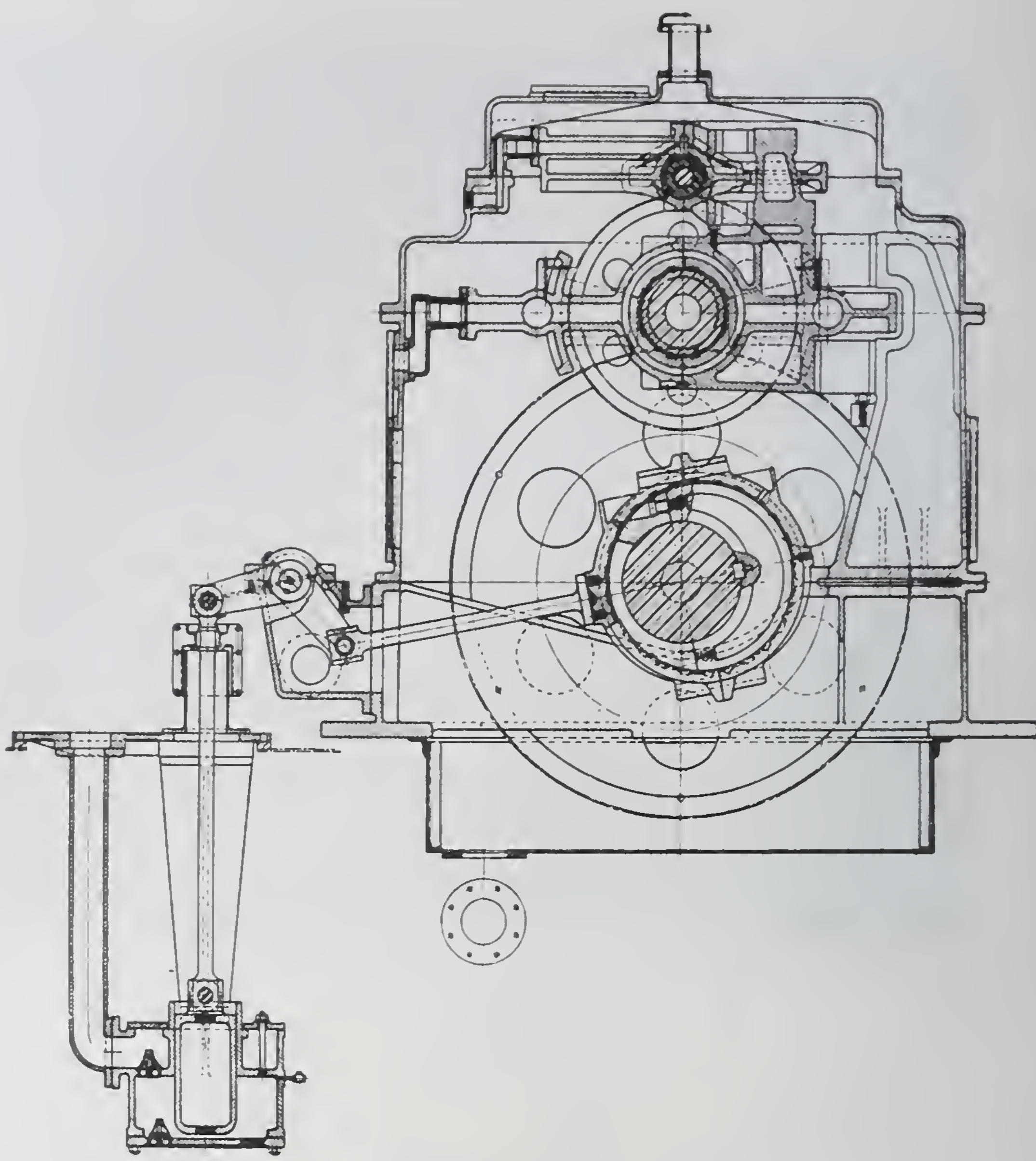


Fig. 20B.

On December 10, 1917, the following statement holds:

TABLE II

Ships	52	U. S. Navy vessels
	4	Foreign Navy vessels
	244	Merchantmen
	300	Total

	Number		Horsepower
Land gears	110	Delivered	121 539
	10	On order	23 577
	120	Total	145 116

Marine gears...	23	Delivered	57 800
	589	On order	2 699 000
	612	Total	2 756 800
Total of above..	133	Delivered	179 339
	599	On order	2 722 577
	732	Grand total	2 901 916*

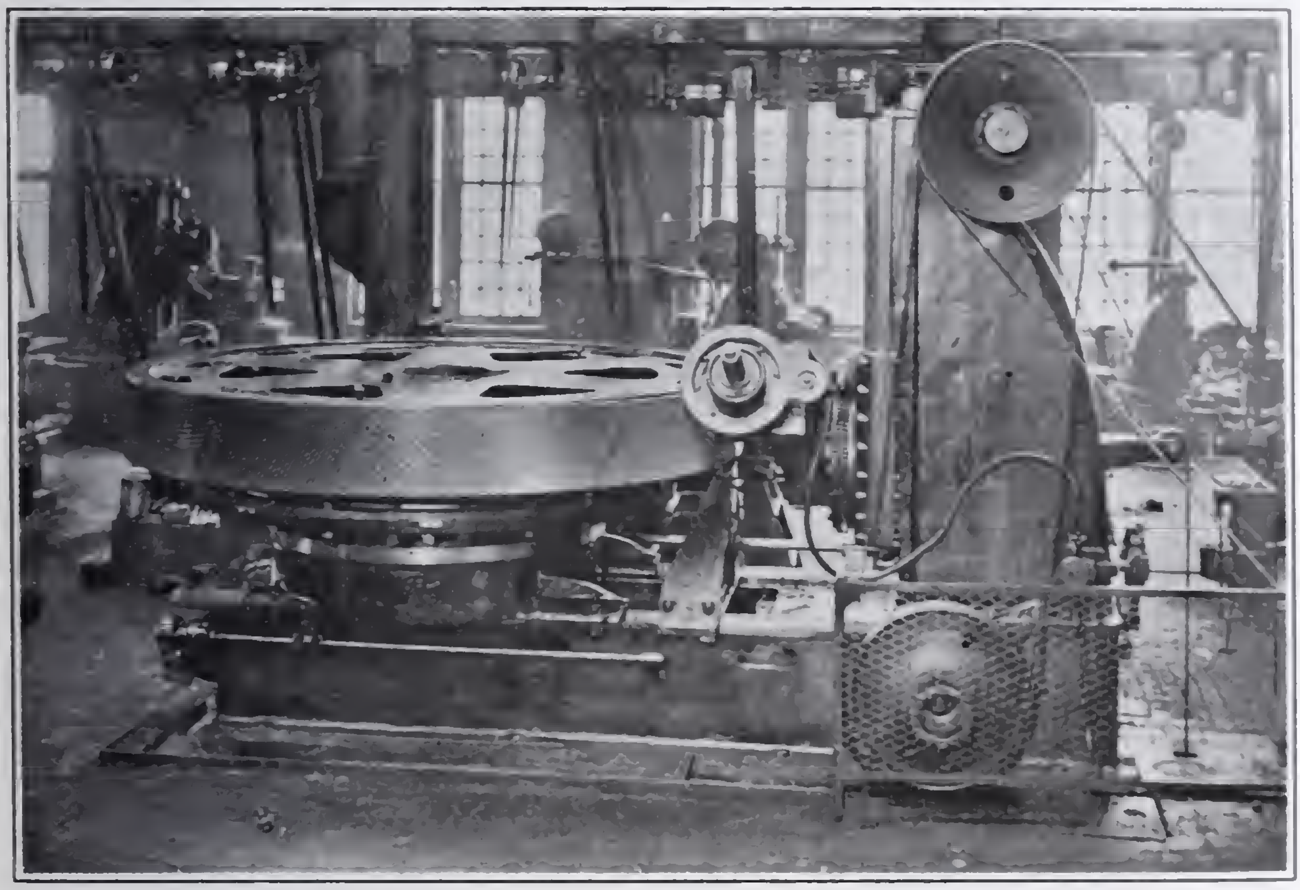


Fig. 21.

The largest naval vessels being engined are three U. S. scout cruisers of 90 000 shaft horsepower each, on four shafts, and one Japanese battleship of 88 000 horsepower. Three of the naval vessels have only cruising gears. With the exception of the *Maui,* all the merchant vessels are single screw and double reduction, having two or in a few cases three gears each.

The gear teeth are cut by hobbing-machines, and the installation of these machines has expanded with the expanding busi-

*On Feb. 1, 1918, this total has risen to 3 117 916 hp. in 792 gears. The number of ships is 327.

ness. Fig. 21 shows the first of these, purchased from Messrs. Schuchardt & Schuette. It is seen cutting the gear (Fig. 17) for a foreign battleship. This machine can cut gears up to 154 inches diameter by 30 inches breadth of face. The installation of these machines now at work includes also three made by the Westinghouse Machine Company—capacity 160 inches diameter by 32 inches face—one 100 inches diameter by 28 inches and one 96 inches diameter by 20 inches, and so on down to the two of smallest size, 24 inches by 10 inches. In all there are fifteen machines. There are also being made by the Westinghouse Machine Company, eight of the present largest size machines, 160 by 32, and six of 72 by 20, and, on order from Messrs. Gould & Eberhardt, four 36 by 14—in all 18 new machines. This makes a total, running and building, of 33 hobbing-machines. Designs for even larger machines than the above have been made.

On account of the expansion of this business it became necessary to greatly enlarge the manufacturing facilities. In November 1916, 500 acres at Essington, on the Delaware River, about eight miles below Philadelphia, were purchased by the Westinghouse Electric and Manufacturing Company which, as is well known, absorbed the Westinghouse Machine Company some considerable time ago. Contracts were let on February 22, 1917, and work on the plant started near the end of March. The part being built now, covers 120 acres of this ground and will cost about $10 000 000. Work was started in the pattern shop at Essington in August, in the foundry in November, and in the machine shop in December. By July 1918 it is hoped to have 6000 men at work and the total capacity of the complete plant will be about 22 000 men. For a considerable time the Pittsburgh works of the Westinghouse Machine Company will continue to manufacture gears and turbines but later all this will be transferred to Essington. A description of this large plant will appear in an early number of the *Electric Journal.*

DISCUSSION

Mr. A. Peterson:* I believe it is not out of place to point out a little more strongly than the author of the paper has done, that the entire credit for the introduction of the modern high-speed helical gearing is due Dr. De Laval. When he started to develop a gear for connection to his high-speed turbine, he was confronted with and had to overcome, and successfully did overcome a great many more difficulties than had to be overcome by those who about 20 years later simply adapted the De Laval helical gearing to large powers both without and with some slight modifications of his original design. The success of the modern high-speed helical gearing of to-day, is the result of Dr. De Laval's pioneer work about 30 years ago and he should be given proper credit for this work.

While it is true that the earlier De Laval gears were built only for comparatively small power, the limit of about 200 horsepower placed by the author is not correct, as a considerable number of gears connected to De Laval single-stage turbines ranging between 500 and 600 horsepower for driving centrifugal pumps and generators are in successful operation to-day. Many of these larger gears have been in continuous operation approximately 10 years without having been replaced, and they operate just as well to-day as on the day they were started. No trouble whatsoever has been experienced with uneven distribution of tooth pressure, due to bending or torsional displacement, or to misalignment after the gear reduction has once been properly aligned. The author is therefore not correct in his belief, that the difficulties in overcoming trouble due to bending or torsional distortion of the pinion, and to misalignment, confined the original De Laval gear to comparatively small powers. The limiting feature was *not* in the gear, but in the turbine which as generally known was of the single-stage type, which type, as recognized by turbine designers, is limited for condensing work to rather small powers. Without going into the reasons for this limitation, I may state that the maximum horsepower for which an economical single-

*Assistant Chief Engineer, De Laval Steam Turbine Company, Trenton, N. J.

stage turbine could be designed, is between 750 and 1000, and in 1909 a design for a 750-horsepower turbine and gear reduction was worked up but never put through the shops, due to the fact that at the same time the De Laval Steam Turbine Company decided to design and build multi-stage impulse turbines as this opened up a wider field without any restrictions as to maximum power.

A gear reduction of the rigid bearing type for transmission of a load of about 1000 horsepower, was designed at the same time on exactly the same lines as the original De Laval gears used in connection with the small size De Laval single-stage turbines—that is a single pinion with two or three bearings meshing into a single gear. Practically no change was made from the original design, except to make the pitch diameter of the pinion larger, due to the great decrease in speed, and to make the teeth of greater pitch.

The rigid-frame type of gear is now used successfully for all kinds of service and at considerable powers, not only as built by the De Laval Steam Turbine Company, but by numerous other concerns, such as for instance, the Parsons Marine Steam Turbine Company in England, this company probably having in successful operation at the present time more *rigid-frame gears* for marine work than all other builders put together.

One of the main reasons for the adoption of the floating frame seems to have been the difficulty in obtaining correct alignment, and to compensate for wear of the bearings. The builders of rigid-frame gears have not experienced any difficulties in this respect, and as far as wear of bearings is concerned, the modern method of forced lubrication giving a perfect oil film between the shaft and the bearing, has made all wear of bearings practically impossible even when operating under very great pressures per square inch. Although the inventors of the floating frame did not have in mind that the floating frame would compensate for bad gear cutting, there was, however, a slightly different understanding regarding this even among employees of the Westinghouse Machine Company.

I quote from page 70 of a paper * by Mr. H. A. Rapelye:

"To review the functions of this floating frame, they are, to compensate for distortion of the pinion, due to the power transmitted, and also to compensate in a large degree for such errors of spacing or helix angle as may occur in the gear due to its large diameter."

It may be interesting to know the opinion regarding the floating frame by such an authority as Sir Charles A. Parsons. I quote from a paper appearing in *Engineering* (London), March 14, 1913, v. 95, p. 372:

"Examination of the teeth of gear-wheels which have been running for some little time, transmitting large powers, shows the work to be distributed over the teeth with fair uniformity, and confirms the opinion expressed by the author before this Institution [the Institution of Naval Architects] in 1910 in his reply to the discussion on the first of the papers above referred to, that with double helical gear such devices as floating frames for the pinions or hydraulic pistons to distribute the load equally over the pinion bearings are totally unnecessary, the natural elasticity of the supporting structures providing all the accommodation necessary—assuming, of course, reasonably accurate alignment of the shafts. The pinions are in all cases connected to their turbine-shafts by flexible couplings, which allow them longitudinal freedom, and this in itself, with double helical gears, ensures that the load is practically equally divided between the right and left-hand portions of the gear.

"Careful investigations have been made of the causes producing noise, with the object of removing such causes and obtaining a silent gear. These investigations show the noise to be due to slight inaccuracies in the teeth, the order of accuracy required for silent gearing being higher than present gear-cutting machines are capable of affording. Fig. 18 [*Engineering* (London), v. 95, p. 373] is reproduced from a photograph of a microphone oscillograph record obtained from a double helical gear-wheel by suspending over the gear case a microphone connected with an oscillograph. It will be observed that definite notes are produced. In the particular case illustrated the frequency was found to be 160 times the number of revolutions of the wheel, and its source was traced to the parent gear of the gear-cutting machine—viz., the single worm and the 160 teeth of the worm-wheel, which rotated the table on which the work was mounted while the wheel was being cut. The inaccuracies of this gear were carefully measured, and found to be co-periodic with the worm-wheel teeth, and to have a double amplitude of about 4/1000 in.

"In the case of the gear wheel referred to above, as there did not appear at the time to be any means of removing the irregularities from the teeth, and very silent running was desired in this instance, stiff springs were fitted above and below the bearings, having a small amount of initial compression and permitting a movement of about 1/100 in.

*Proceedings of the Association of Iron & Steel Electrical Engineers, 1913, p. 51.

as the load was increased to its full value. The pinions being thus flexibly supported, noise and shock were to some considerable extent intercepted, instead of being transmitted to the structure of the gear-case. It was recognized, however, that spring supports were an imperfect remedy, the real remedy being a higher degree of accuracy in the teeth."

If the claims—that two to three times greater tooth pressures, higher pitch-line speeds and quieter operation were possible by the introduction of the floating frame—were substantiated by facts and by actual experience of the Westinghouse Machine Company, there would of course be an excuse for going to such complications as are involved by the introduction of the floating frame. The following facts may be interesting:

The tooth pressures used by the Westinghouse Machine Company on floating-frame gears on which data are available do not seem to be a whole lot greater than used for rigid gears. The pitch-line speeds are *not* greater. There are many rigid-frame gears operating at pitch-line speeds greater than 6000 feet per minute and they are operating just as quietly as any floating-frame gear.

A great point is being made of the fact that a considerably higher so-called "power constant" is used for floating-frame gears. This does not mean a great deal, unless the width face of the gearing is given so that the tooth pressure per inch face can be calculated.

The tooth pressure per inch face together with the ratio of width face and diameter of the pinion, are the principal factors governing the design of helical gears. A high "power constant" means only that for a certain speed and horsepower a comparatively small diameter pinion is used, and this does not necessarily result in any considerable saving in weight or space.

If the tooth pressure per inch face is fixed and a certain speed and horsepower given, a diameter of the pinion (small diameter and high "power contsant") will result in a width of face $= 1$. Increasing the diameter of the pinion to $K \times d$ will result in a narrower face $= 1/k$ for the same tooth pressure. The difference in weight between the two gear reductions will not be great as the weight of the two large gear rims, one with small diameter and wide face, and the other one with larger diameter and nar-

rower face, will be exactly the same for the same thickness. There will of course be a slight increase in weight for the larger diameter gear reduction, due to longer spokes and slightly larger gear casing. On the other hand, as can be seen from cuts, the introduction of the floating frame causes quite an increase in weight and also in overall width of the gear casing.

There is no reason why a manufacturer of rigid gears could not use the same diameter of pinion as used in connection with floating-frame gears, but the larger diameter pinion has many advantages, such as a greater number of teeth for the same pitch, and this will give quieter operation.

As to the tremendous tooth pressures (two to three times greater than are possible with rigid gears) claimed to be used in connection with floating-frame gears, the following may be of interest:

Seven gear reductions for driving generators, described in a paper by Mr. H. T. Herr, Vice President of the Westinghouse Machine Company (Journal of the Franklin Institute, March 1913, v. 175), operate at maximum rating with tooth pressures ranging from 365 pounds to 455 pounds per inch face. The speed of the pinion in all cases was 3600 r. p. m. and the diameters of the pinions were approximately the same as would be used by builders of rigid-frame gears. One of these gear reductions is the Commonwealth Steel Company installation referred to by the author, and is rated at 1000 horsepower, maximum, under which conditions the tooth pressure is 365 pounds per inch face.

A floating-frame gear reduction offered on a municipal pumping job rated at 1300 horsepower at 3600 r. p. m. had a width of face which gave 360 pounds tooth pressure per inch with a pinion diameter of 7½ inches. The rigid-frame gear reduction which is at present installed at this place has been in continuous operation for about three years, 24 hours per day, with a tooth pressure of approximately 375 pounds per inch face. The speed of the pinion is 3600 r. p. m. and the diameter 6.6 inches.

The above data are taken from the earlier floating-frame gear reductions and the claim may be made that high pressures

were not used at this time. However, the following will show that even the latest high-speed floating-frame gear reductions are not designed for very high tooth pressures.

An article in *International Marine Engineering,* August 1916, v. 21, p. 359, describes the propelling machinery for a great number of cargo steamers equipped with floating-frame gear reductions. The *Malmanger* referred to by the author is apparently one of these steamers and it must be taken for granted that in order to save space and weight the tooth pressures used are as high as considered safe for this particular kind of service. Although the dimensions of pinions and gears are not given, the drawings seem to be made to scale and with the dimensions given, such as center distances and speed ratios it is possible to obtain very closely the principal dimensions, from which the tooth pressures can be calculated. The high-speed pinion is approximately 5.15 inches in diameter and the width face approximately 20 inches, giving for normal load of 1250 horsepower per pinion, a tooth pressure of approximately 430 pounds per inch face and for maximum load of 1450 horsepower a tooth pressure of approximately 465 pounds. These pressures are *not* higher than used by the De Laval Steam Turbine Company on marine gears of the rigid type and are almost exactly the same as used by Parsons for the steamers *Ciudad de Buenos Aires* and *Ciudad de Monte Video* referred to by the author.

The diameter of the low-speed pinion is approximately 10.25 inches, and the width of face approximately 40 inches, giving for normal load a tooth pressure of approximately 855 pounds, and for maximum load 925 pounds per inch face, which figures are not higher than common practice for low-speed pinions. Since 1912 there has been in continuous successful operation at Ross Pumping Station, Pittsburgh, a 600-horsepower De Laval gear reduction, the pinion speed being 600 r. p. m., diameter 6.6 inches and width of face 20 inches, giving a tooth pressure of 955 pounds per inch, and such tooth pressures are used on many low-speed De Laval gears.

The weight of the 2500-horsepower marine gear reduction described in *International Marine Engineering* is given as 34 tons complete, including two gear-type oil pumps driven by means of

spiral gears from the first gear reduction. The weight of a De Laval 2800-horsepower rigid-frame marine gear reduction including a first and a second gear reduction, but excluding oil pumps, is 32 tons. This gear reduction has a maximum rating of 3000 horsepower. This shows conclusively that the big saving in weight claimed to be obtained by the introduction of the floating frame and by use of a high "power constant" does not materialize in actual practice. By the above I have tried to show that the high tooth pressures made possible by the introduction of the floating frame are more imaginary than real, and also that the value of a high "power constant" is rather doubtful.

The experimental gear referred to by the author, was tested for 40 hours at approximately 6000 horsepower, which corresponds to a tooth pressure of approximately 900 pounds per inch face. The author states "We know now that we could with safety have loaded it at least three times as much". This would have meant a tooth pressure of 2700 pounds per inch face. I question whether anybody has had any experience with tooth pressure of such magnitude in connection with high-speed helical gears. There are probably very few, if any, *high-speed* gears of this type operating *continuously* at tooth pressures even as high as 900 pounds.

In connection with certain kinds of work such as, for instance, some classes of war vessels, which operate only for comparatively short times at maximum power, high tooth pressures are being used. Also in generator work where the load is fluctuating high tooth pressures may be used, but when it comes to practically continuous operation at full load, such as occurs in merchant marine work, the tooth pressures on the high-speed gear reductions are brought down, and this practice is followed not only by builders of rigid-frame gears, but also by the manufactures of the floating-frame type in spite of statements to the contrary.

As a matter of information and also to show that the ship builders have not lost their faith in the rigid-frame type of gear reduction, I may state that the De Laval Steam Turbine Company has on order at the present time a total of approximately 2 000 000 horsepower in gear reductions.

MR. THOMAS FAWCUS:* I have read the paper with great pleasure and interest, and do not feel that I could in any way criticize the conclusions of my friend, Mr. Macalpine. He has given this device years of thought, and it is without doubt a very valuable mechanism.

My personal efforts have been directed more towards the perfecting of machines for accurately and economically making the double-helical type of gear for general commercial purposes, and the only turbine transmissions manufactured by the Company which I represent have been of small sizes, below 1000 horsepower, the gears having comparatively narrow faces, and requiring only two pinion bearings. These have all been of the so-called "rigid" type, as we have found little difficulty in securing and maintaining correct alignment. Next to accurate tooth cutting, the importance of accurate alignment is self-evident, and any device which will automatically and constantly secure this must be of great value especially where high speeds are involved. It has taken some little time to educate gear users to this fact. I remember some four or five years ago we furnished a pair of fairly large gears for a mill drive, and a complaint was made that they did not operate properly. We found upon examination that the pinion shaft was out of line 3/16 inch in about 4 feet. Upon pointing this out we were told that if herringbone gears would not work properly with so slight an error in alignment, they were no good. I am glad to say that troubles of this kind are rapidly becoming a thing of the past.

I would like to ask Mr. Macalpine if he can tell us whether the "floating frame" or any similar device is used by manufacturers of turbine gears abroad, and if not, how their reduction gears compare at the present time with those containing the "floating frame". That Mr. Macalpine's estimate of the "floating frame" is not universally accepted, I mention merely in order to promote some discussion of the subject.

I have before me a paper on "Mechanical Gearing for the Propulsion of Ships", by Sir Charles A. Parsons, read in March 1913, before the Institution of Naval Architects.† He states that at that time there were under construction turbine machinery and

*President, Fawcus Machine Co., Pittsburgh.
†Transactions of the Institution of Naval Architects, v. 55, pt. 1, p. 48.

mechanical gearing representing a transmission of over 120 000 horsepower. Some time after this, I might add, young Mr. Parsons informed me that the firm of C. A. Parsons & Sons Co. had orders for about 300 000 horsepower. All of these transmissions, I believe, were of the "rigid" bearing type. In the above paper, Sir Charles says:*

"Examination of the teeth of gear wheels which have been running for some little time, transmitting large powers, shows the work to be distributed over the teeth with fair uniformity, and confirms the opinion expressed by the author before this Institution in 1910 in his reply to the discussion on the first of the papers above referred to, that with double helical gear such devices as floating frames for the pinions or hydraulic pistons to distribute the load equally over the pinion bearings are totally unnecessary, the natural elasticity of the supporting structures providing all the accommodation necessary, assuming, of course, reasonably accurate alignment of the shafts. The pinions are in all cases connected to their turbine shafts by flexible couplings, which allow them longitudinal freedom, and this in itself, with double helical gears ensures that the load is practically equally divided between the right and left-hand portions of the gear."

In the discussion which followed, Mr. S. Z. de Ferranti said:

"A little later on I was very much interested in the experiments of Mr. George Westinghouse, carried out on a considerable scale at Pittsburgh, with a reduction gearing for lowering the turbine speed as applied to propellers. In looking into the matter at that time I came to the conclusion that everything would turn upon the accuracy of the gear.

"I did not like the ideas of Mr. Westinghouse for flexibly supporting the pinions or doing anything of that kind to get over difficulties which were the result of some mechanical imperfection, so I determined to cut some wheels in order to try what could be done in the way of accuracy. We had a Schuchardt & Schutte machine at the Vickers' works in Sheffield, and I determined to try some wheels cut on this machine and see how far they were accurate. For this purpose I made some very carefully got-up blanks, ground the large diameter mandrill on which they were fixed on the gear cutter, put two blanks one on top of the other, and then cut them on the machine. This showed, when I turned one blank round to 180 deg. on its mandrill exactly what the error was. It doubled the error when the teeth were put together at one side."

Mr. Ferranti then proceeded to describe a machine very similar to that used by Sir Charles Parsons for overcoming this

*This quotation, appearing elsewhere, is from a reprint in "Engineering;" hence the variations in punctuation.—Ed.

error. Both of these gentlemen, you will note, at that time considered accuracy of cutting the important point, and seem to have placed little stress on extreme delicacy of alignment. Of course this paper was read four years ago, and it would be interesting to know if the opinions of the gentlemen mentioned have undergone any modifications since that time, in favor of the "floating frame" or any similar device.

An interesting side-light on the double-helical reduction gear is the demand of recent years for similar gears for driving mills and machines from electric motors, etc. The merits of the herringbone type of gear are well known, but the difficulty of making the gears at a reasonable price and with reasonable accuracy had to be overcome. There is considerable difficulty in making up a large herringbone gear of two separately cut right- and left-hand helical gears, besides which it is an expensive process. To overcome this, machines have been developed which cut the two sides of the gear (right- and left-hand) simultaneously. In this way very accurate gears are easily constructed at a comparatively low cost. The Fawcus Machine Company has now an equipment consisting of fourteen of these special machines, each one using two hobs or cutters simultaneously, and consequently being approximately equivalent in productive capacity to twice that number of single cutter machines. Nine of these machines will cut 120-inch diameter, one 180, one 240, and the remainder 72-inch and under.

Mr. Percy C. Day:* Mr. Macalpine's paper is practically confined to an exposition of the virtues of the floating frame. Now there are a good many people who have assisted in developing herringbone gears along rigid lines, and I would like to contribute a few words, not in criticism of the floating frame, but merely to show that successful marine installations have been built in large quantities without disclosing any vital need for this special device. Mr. Macalpine paints a lurid picture of the evils due to cross-bending, torsion and misalignment through wear of bearings or distortion of gear frames. I think that if these difficulties had been of such a pronounced nature as Mr. Macalpine

*Manager, Gear Department, Falk Co., Milwaukee.

suggests, the rigid frame would have died at birth and the great number of successful installations which have been made along rigid lines would never have materialized.

Perhaps the first successful geared marine drive on a practical scale was the installation in the S. S. *Vespasian*. The *Vespasian* gears were made by the Parsons Marine Steam Turbine Company. I was connected with the Power Plant Company, Ltd., at the time and was responsible for the choice of dimensions for these gears, which were as follows:

Horsepower—1000 (500 hp. per pinion)

R. p. m.—1500 to 75

Pitch diameters 5″ and 100″

Pinions, 20 teeth, 4 d.p., 23° spiral angle

Total width of face 20″ (10″ on each side of center pinion bearing)

Materials:

Pinion—chrome nickel steel.

Gear—mild steel rim.

It should be noted that the length of each helix is twice the pitch diameter, which is the proportion Mr. Macalpine recommends as the best. The so-called power constant is 2.67. At the time when these gears were designed very little was known about tooth-load capacities, consequently everyone connected with this job wanted to play very safe.

The *Vespasian* was an old vessel at the time when the geared turbines were installed (about 1909), some eight years ago, and I have information that the hull was recently condemned but that the turbines and gears were in such good condition that they were taken out and put in a new ship. Many ships have been built with geared turbines since the *Vespasian* demonstrated the efficiency of this method.

Mr. Macalpine gives us some figures which show that a large number of vessels are being built in this country with floating-frame gears. He does not give us any information in regard to the number and total horsepower of the rigid-frame drives that have been built in England alone during the last eight years, but there is no doubt that the number is very large indeed and that warships, passenger ships and freighters are included.

Everyone in America who can build marine gears or turbines is busy at the present time. At this date the Falk Company has completed marine-turbine drives totalling approximately 150 000 horsepower and has also cut marine gears for other concerns for about 80 000 horsepower. It has under construction at the present time, gear units totalling nearly 3 000 000 horsepower. All these gears are of the rigid type.

It is claimed for the floating frame that it allows the use of smaller pinion diameters and smaller gear dimensions to transmit a given power and speed than is possible with gears mounted rigidly. It is true that the advocates of the floating frame appear to use smaller dimensions than are considered desirable for rigid gears, but this may be merely a question of optimism, since there appears to be no positive evidence to show that one gear will transmit substantially more than the other.

The questions of cross-bending and torsion in the pinions are largely matters of design. If we accept Mr. Macalpine's statement that the best proportion of pinion has a length of each helix equal to twice the diameter, then there is a definite relationship between the pitch diameter, the speed and the horsepower transmitted. This relationship is demonstrated by Mr. Macalpine and called the power constant. It should be noted, however, that the power constant becomes misleading directly we change the relationship between the width and diameter of the pinion. Now in many cases the pinion diameters have to be settled by questions of available space and clearance. This is particularly true in the case of high-power gears for high-speed vessels where the speed reduction is relatively low and where two turbines, with one pinion each, drive into the same gear. It frequently happens that the gears have to be made larger in diameter than would otherwise be necessary in order to provide clearance for the turbines and steam-pipes alongside of each other.

Mr. Macalpine lays great stress on cross-bending, torsion and misalignment as a means of concentrating the load in spots on the gears. I think that this load concentration may just as easily be caused by inaccuracies in gear-cutting and when it is so caused the floating frame does not help to cure the trouble. Much can be done by increasing the accuracy of the methods

used for cutting the teeth and when gears are cut with extreme accuracy, and carefully aligned in rigid frames, there is no difficulty in making them bear evenly along the entire length of face, provided they are so designed as to practically eliminate the effects of cross-bending and torsion. Cross-bending and torsion can be eliminated by merely increasing the diameter of the pinion in relation to its length. Such an increase in pinion diameter increases the pitch-line velocity of the gears and calls for a higher degree of accuracy in the gear-cutting. Gear designs have been developed along these lines in connection with double-reduction marine drives with the result that it is possible to produce a very rigid and substantial gear unit which involves no more weight or space than a drive with floating frame of similar capacity using smaller gears and higher tooth loads. It is not at all a difficult matter to scrape in the bearings on a gear unit so that the shafts are in accurate alignment and there is no reason why a gear unit of this type should cost any more to build, or weigh more, than a floating-frame unit of similar capacity even when we concede that the floating frame justifies smaller dimensions for the gears and pinions. I believe that the weight of the *Malmanger* gears was 85 000 pounds for 2900 horsepower. The Falk Company is building rigid gears of about the same horsepower and speed reduction with a net weight of 65 000 pounds.

Gear drives for marine turbines must be separated broadly into two classes. For a merchant ship a gear must be designed to carry its full load for weeks at a time with little variation and it seldom happens that the gear has to run for long intervals at reduced load. The propeller speeds of merchant ships usually lie between 75 and 100 r. p. m. and turbine speeds usually do not exceed 1500 to 2000 r. p. m. with single-reduction gears. Double-reduction gears have been introduced for marine drives only within the last two or three years and this change has had the effect of allowing an increase of turbine speed up to 3000 or 4000 r. p. m. with a gear reduction of 30 or 40 to 1. The total weight of turbines and gears is so much less than the weight of the reciprocating engines displaced that it has not been considered necessary to cut down the weight of gear equipment to the last pound and designs have been developed along substantial lines.

Gear units for merchant-marine work are usually built with substantial frames of cast-iron supplemented by covers and pans of the same material. There is little if any risk of sufficient distortion of the bearings to disturb the alignment of the gear teeth. There is no reason why liberal bearing surfaces should not be provided and if lubrication is properly taken care of there should be no appreciable wear in the bearings. As a natural result of running for long periods under constant load the teeth wear themselves into perfect alignment. If the bearings are carefully proportioned to give equality of load on all journals in one line, then a certain amount of wear can be permitted without risk of change in alignment.

Gears used on war-ships must be designed for totally different conditions. Generally speaking, these gears will run at less than half load for most of the time and full load is required only during occasional bursts of speed which are of relatively short duration. Reduction of weight is of first importance, but it is also desirable to eliminate cross-bending and torsion as much as possible because these effects vary with the load and the gears have to run under very variable load conditions. On the other hand the maximum load is never sustained for very long at a time and higher tooth pressures can be used on a maximum load basis. In most cases the propeller speeds are high with this class of gearing and the horsepower transmitted is much greater than for cargo ships, hence the average turbine speed is inclined to be lower, and single-reduction gears are used in nearly all cases.

Quite recently I had an opportunity to compare the essential dimensions and weights of gear units with rigid and with floating frames, designed for practically identical conditions. Each unit has to transmit several thousand horsepower at very high speed and is of the single-reduction type with two turbines and two pinions meshing with the main gear. In both cases the diameters and velocities of the gears have been chosen from considerations of available space between the two turbines. I do not feel myself at liberty to give actual figures but the relative figures may be interesting. In each case the lower figure is given as unity so that comparison can be easily made:

TABLE III

	Rigid frame	Floating frame
Tooth pressure per inch of face..	1	1.36
Pitch-line velocity	1.11	1
Pinion journal velocity..........	1	1
Pinion bearing pressure per sq. in.	1	2.2
Metal thickness in frame........	1.33	1
Depth of frame................	1.71	1
Metal thickness in gear spider....	1.33	1
Pinion shaft diameter...........	1	1
Main-shaft journal diameter.....	1.23	1
Length of main bearings........	1.3	1
Power constant	1.6	2.15
Thickness of sheet-metal covers..	1.5	1
Net weight of gear unit..........	1.3	1

From the above comparison, it will be noted that the rigid drive weighs 30 per cent. more than the floating-frame drive and that the tooth pressures per inch of face are in about the same proportion. If we look closely into the figures, however, it will appear that the lighter weight of the floating-frame design is not necessarily due to the difference in gear face but is accounted for by the lower frame, thinner metal section and greatly reduced bearing area for the pinions, combined with much smaller dimensions for the main shaft and bearings. None of these lightened features appear to bear any relationship to the floating frame itself and it would only be necessary for the designers of the rigid frame to accept their competitors' figures in regard to shaft and bearing sizes, and prudent frame dimensions, in order to reduce the weight of the rigid drive to substantially the same as that of the floating frame and still provide the increased gear face which is included in the above comparison. In other words, if we admit that the floating frame permits an increase in tooth pressure per inch of face with smaller and lighter gears, it seems that the weight saved has to be put back into the floating frames and there is little, if any, net gain in weight or dimensions.

There are other and more practical questions which should be considered. If anything happens to the bearings while a ship is at sea, it is of utmost importance that repairs and replacements can be quickly made. Marine gear units with rigid frames can very easily be designed so that all the bearing caps can be lifted and bearing shells replaced without removing shafts, gears or covers. The floating-frame design makes this kind of construction difficult and the bearings are not readily accessible in any of the examples I have seen.

The idea of measuring the weight and cost efficiency of herringbone-gear drives by means of Mr. Macalpine's power constant would be very misleading if generally applied. The following are two extreme examples taken at random from gears which have been constructed recently. A standard Falk S-60 gear unit is used for driving a hot mill from an electric motor. The maximum load transmitted is 7000 horsepower at 270 r. p. m. of the pinion. The pinion is 20-inch pitch diameter, 40-inch face, and has two bearings. The power constant for this pinion appears to be 3.24. Now it should be noted that the pinion has a length of two diameters and by substituting a pinion with three bearings it could be made with a net face of 80 inches without violating Mr. Macalpine's rules for best proportions. This would increase the maximum load capacity to 14 000 horsepower and the power constant would be increased to 6.48. The other extreme is a new power-house unit built by the Falk Company for its own use with a view to throwing some light on the question of permissible pitch-line velocity. The pitch diameter of the pinion is 11.8 inches and the velocity is about 11 100 feet per minute. This gear is designed for 1850 horsepower, with pinion speed of 3600 r. p. m. The gear face is proportionately narrow with the result that cross-bending and torsion are practically non-existent. The design is quite compact for the power transmitted and the unit compares very favorably as regards weight and cost with a normal design for a gear unit of this size. But it should be noted that the power constant figured in accordance with Mr. Macalpine's formula is only 0.3 or considerably less than 10 per cent. of the average power constant of gears described in his paper.

If a formula of this kind is to be used for comparison of gear units of different type, then it must necessarily include the relative gear face as one of the variables or the results obtained will be entirely misleading.

Author's Closure: Before replying to the discussion on my paper I must express my deep regret at the untimely death of Mr. Thomas Fawcus. I had not met him often but from the first I was struck not only by his open and manly courtesy but by his deep interest in his profession. I am given to understand that it was by hard and unremitting efforts that he raised himself to a position of great usefulness at the head of an important and growing company. He had shown me the machines he had built and was building to attain, and which did attain, great accuracy in the cutting of gears and to accomplish this expeditiously and economically. He very properly placed accuracy as the one absolute requisite of a successful high-speed gear. I have no doubt that his company by its success, following the impetus and direction he has given, will long keep his name prominently before the engineering profession. I shall always be grateful for the kindly appreciation he has expressed here of the value of the floating frame.

Replying to his questions: Floating-frame gears have so far, been built only by the Westinghouse Machine Company. In 1914 two licenses were granted in Europe but the war has prevented any work being done under them. A very favorable comparison with the rigid gear is made by Mr. Alexander Cleghorn, of the noted Fairfield Shipbuilding and Engineering Company, in his recent presidential address before the Institution of Engineers and Shipbuilders in Scotland, and the floating frame has many friends in Britain, but I am not aware that Sir Charles A. Parsons and Mr. Ferranti have yet seen the light. I will refer to their criticism somewhat later.

I may add that the Japanese Government is arranging a license agreement to build our gears. They have much experience with rigid gears and their action was taken after long and careful investigations of our construction, and practical results.

I am very much pleased that my paper has evoked so much discussion from men of large experience in building rigid gears.

The differences of opinion are curious and instructive. Mr. Fawcus thought that any device which will automatically and constantly secure good alignment must be of great value. Mr. Peterson has experienced no trouble from cross-bending, torsion, or misalignment. The last-mentioned is prevented, according to the advertisements of his firm, by the housing "holding pinion and gear shafts *rigidly* in the same plane"—I presume the meaning is that the axes are held parallel—yet a quotation is given, with apparent approval, from Sir Charles A. Parsons, whose pinions are brought into alignment by "the natural *elasticity* of the supporting structures providing all the accommodation necessary, assuming, of course, *reasonably accurate* alignment of the shafts." Mr. Day thinks that the difficulty is not of a "pronounced nature", and that if the axes are "carefully aligned in *rigid* frames there is *no* difficulty"; but that "as a natural result of running for long periods under constant load the teeth *wear themselves* into perfect alignment." His opinions, put in juxtaposition here, seem a little contradictory. What happens during the long period while the teeth are wearing themselves true? Our experience is, as I have shown, that teeth bearing well do not wear sensibly, consequently the wearing-in period would be exceedingly long if the teeth were not abrading. (The italics in this paragraph are mine.)

Mr. Peterson's two opening paragraphs are inaccurate. The development of helical gearing was gradual and dates back to that distinguished genius of the seventeenth century, Dr. Robert Hooke (1635-1703), who invented single-helical gearing. Regarding double-helical gearing I will quote from Suplee's English translation (1893) of Reuleaux's "Constructor", from the fourth enlarged German edition of 1889, the year in which Dr. De Laval introduced his geared turbine:

"An objection to the use of spiral gears is the axial pressure K, this, however, can be eliminated by the use of double gears of opposite inclination. Such gears have been known for a long time (White 1808) and for moderate service, have frequently been used, as in spinning machinery, tower clocks, &c., and more recently they have been applied to heavy work, notably for rolling mill gearing, both in Germany and America" (page 141).

Two figures are given; one with the gear-wheel in a single piece; one with the gear-wheel in two pieces, one for each helix. As one example of their adoption here, the American Steel and Wire Company put them into their Garrett mill in 1885 and they ran at pitch-line speeds up to 1440 feet per minute. Even staggered double-helical gears, known commonly now as Wuest gears, to which some virtue is supposed to attach, had developed by 1886. (See George H. Reynolds's United States patent, no. 342134.) All honor to Dr. De Laval for the beautiful and accurate work he did, but especially for his turbine. In the single gear and pinion he merely took what was then a well-known device, probably forming the teeth with more care than had previously been done, though the foregoing example at 1440 feet per minute must have been very accurate. (I have made only the slightest inquiry and this may be far from the greatest speed in use at that time.) Thus Mr. Peterson's reference to a double-helical gear and pinion as "De Laval helical gearing" is without any justification. The only gearing which properly bears this name is one in which a pinion is placed between two gear-wheels, and in adopting this arrangement I think Dr. De Laval made a serious mistake which probably delayed for many years the recent great development of reduction gearing. I will justify my opinion a little later. I can well recollect that the De Laval turbine aroused the greatest interest in Europe and America and great expectations for it were entertained. These were not fully realized and it took its place among other motors for small powers, one factory of moderate dimensions being sufficient for its production in the United States. The De Laval gear is entirely unsuitable for marine application (though Sir Charles A. Parsons applied it to a small launch) for it is altogether forbidding to have both propellers, in a twin-screw ship, driven by one turbine and pinion. For safety and maneuvering they must be made independent.

Thus no advance was being made in the introduction of the geared turbine in the first years of the twentieth century. Only small powers were being manufactured and Parsons had wholly given up hope of a development of the turbine through gearing. (See *Transactions* of the Institution of Engineers and Shipbuild-

ers in Scotland, v. 44, p. 216.) The first drawing of the floating frame shown in my paper is dated September 26, 1906, and I am quite confident that this will come to be generally recognized as the first fixed date in the almost unparalleled expansion which we now see. After we had applied for our patents we made no secret of the device and what we were doing was well known both in this country and Britain and, as I can prove, especially so in Newcastle-on-Tyne. But the first great impulse to this expansion was given when, rising from the small powers of the De Laval turbine, 6000 horsepower at a pitch-line speed of 5500 feet per minute, was easily run off by our experimental gear. This riveted attention all over the world. The view expressed by *Engineering News,* December 30, 1909, v. 62, p. 730 (I commend both articles on our gear in that issue to Mr. Peterson's attention), is, "It seems probable that the writers of engineering history, looking back a few years hence, will set down the year 1909 as notable among other things for the development of a practical speed-reducing gear for use with marine steam turbines." The writer of that article did not think, with Mr. Peterson, that we "simply adapted the De Laval helical gearing to large powers . . . with some slight modifications of his original design," for he continues (p. 731), "The ingenuity and the notable achievement in mechanical engineering is in planning the details of construction of these gears in such a manner as to overcome the defects and hindrances which have hitherto prevented the use of gear wheels for any such speeds and any such large powers as are here transmitted." Before making our rigid-gear design, prior to the invention of the floating frame, the tooth pressures per inch in use for steel gears of small length of face in proportion to diameter was ascertained from Mr. Hugo Bilgram of Philadelphia, and the intensity of pressure on turbine bearings was given to us by the Westinghouse Machine Company. Treating both these conservatively and making the ratio of helix length to diameter such that the cross-bending and torsional errors would probably be negligible, we determined the dimensions of the pinion and gear-wheel. None of the dimensions was changed when the floating frame was added. Dr. De Laval's work had no influence whatever on us, and the statement which Mr. Peter-

son says he wishes "to point out a little more strongly" than I did is not remotely suggested in my paper and is not true. It is true that after our rigid design was made we did get the data of a 300-horsepower De Laval gear and compared the speeds, pressures, elasticities, etc., of the two designs carefully, but modified nothing, as our design seemed superior in every respect.

I will now discuss the question of the power constant and tooth pressure. Messrs. Peterson and Day apparently prefer to use tooth pressure and speak of the "so-called" power constant. I believed that the power constant or some equivalent form was in general use. Mr. Fawcus, with whom I discussed the matter while he was still able to take a lively interest in engineering, considered that in some form it was necessary to intelligent design and this is certainly the opinion of the engineers of the Westinghouse Machine Company. In my article "On Reduction Gears," in *Engineering* (London), v. 101, p. 415, I deduced it from three general propositions which take account of elasticity as well as stress in structures. These propositions prove that in any type of design of reduction gear there is one set of proportions which is better than any other and that in similar gears the safe tooth pressure rises in proportion to the pinion diameter. The substance of these propositions has, I believe, long been well known. Though I do not recall seeing the power constant before I deduced it, I believe my only contribution consisted in giving it a name. In *Engineering* (London), v. 102, p. 527, I

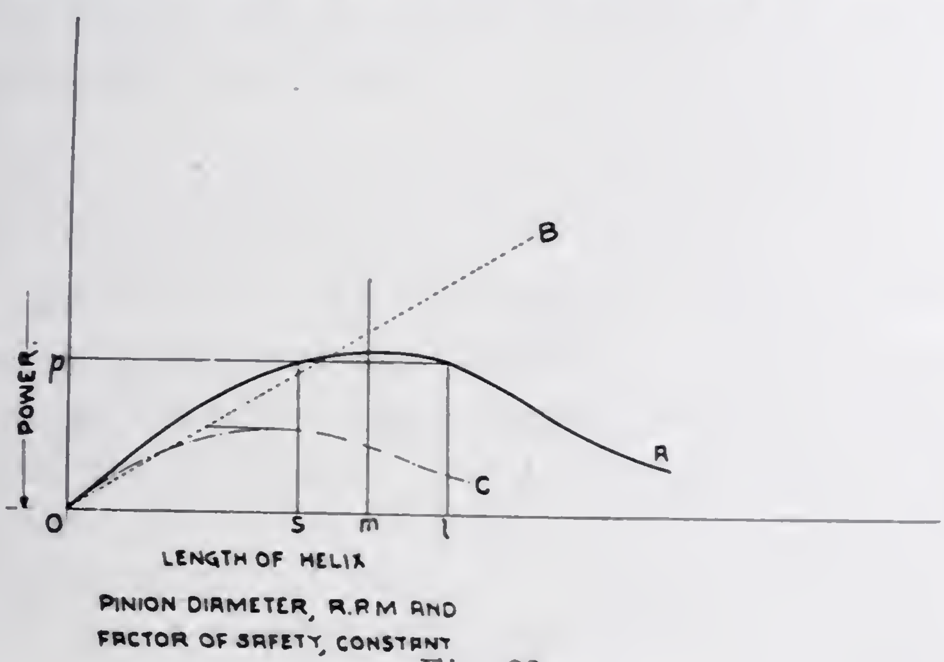


Fig. 22.

have shown that the rise of tooth pressure per inch in proportion to pinion diameter, the number of teeth being constant, also follows, at least approximately, if lubrication is taken into account. Mr. Peterson (p. 42 above), seems first to settle his tooth pressure, then the diameter of the pinion, and last the length of tooth face—apparently treating tooth pressure as independent of diameter; though it is quite easily proved, as just stated, that for equally well-designed gears the tooth pressure rises in proportion to the pinion diameter. It would also follow from the calculation he indicates that the power is proportioned to the length of helix as shown by O B, Fig. 22. Quite obviously, as the length of helix increases, the factor of safety being kept constant, the power follows some such curved line as O A, rising to a maximum for a helix length O m. As the helix becomes still longer the tooth pressure will inevitably become unevenly distributed, since torsion will increase and deflection increase still more rapidly, as can be proved by a simple calculation. Hence the power for a given factor of safety will rapidly fall toward A. I have discussed this subject very fully in my above-mentioned article "On Reduction Gears" to which I refer the reader. From the results of calculations there given it cannot be doubted that the length O m of each helix is not much if any greater than two pinion diameters in a three-bearing pinion. Thus approximately this ratio must be chosen if it is the object to get the most power safely from a given pinion diameter. In the vast majority of turbine reduction gears this is the object and it is to be noted that, of the examples given in this discussion and reply, only two fall considerably short of this ratio.

If in two gear designs the helix lengths are made O s and O l respectively, the power, for a constant factor of safety, is the same and differs little from the maximum at m. Thus the power constant method is an excellent approximation—far within the limits of error of the data we possess. The average tooth pressures are in the proportion of O l: O s, and calculations based on tooth pressure are apt to mislead; or, at least, the corresponding values of tooth pressure and diameter can be arrived at only by trial and error—in spite of Mr. Peterson's indication that it is a single step. Hence for almost all such reduction gears as we

are considering, I maintain that Messrs. Peterson and Day are in error when they think that the power constant is misleading or that it means little unless the ratio of helix length to diameter is given.

In a small proportion of cases, expecially where there are two pinions driving one gear, the turbine diameters require the pinion centers to be farther apart than they would otherwise be made, as Mr. Day states. The pinions then must be of larger diameter and the helix length can be considerably shortened. If, for instance, each helix length is less than 1.5 diameters, a considerable reduction of the power constant from that used for 2 diameters will naturally be made.

While the exact determination of the elasticity of the pinion is a most complex problem, good approximations can be made. Consideration of the great stiffness of the teeth alone shows that misalignment of even 1/1000 inch measured outside the helixes, even in a large gear, is far from an unimportant quantity. Hence if O A represents the result for perfect alignment, the slightest change will cause concentration of tooth pressure, greatly reducing the ordinates, for the same factor of safety, and the point of maximum will move toward O. Such a result as is illustrated by the dotted curve O C obtains.

We can now discuss more clearly whether or not I am correct in believing that there are limitations in the De Laval gear which discouraged its application to large powers. The De Laval gear which I used for comparison with our oirginal rigid-gear design had the following data:

Horsepower	300	
Pinion, pitch diameter............	2.542	inches
Pinion, r. p. m....................	10 510	
Gears, diameter	29.684	inches
Length of each helix..............	9.5	inches

There were 31 teeth in each pinion helix and the helical angle was 45 degrees. The pinion was between two gear-wheels and thus the axial length of tooth contact was 38 inches. If the condition that equal powers be taken from each gear-wheel could be fulfilled with certainty, cross-bending would be done away with and the power constant should be very high. It should be con-

siderably more than double that of the same pinion driving one gear, as this would be subject to cross-bending and have a tooth face of only 19 inches.

We find the following:

Pitch-line speed	6994 ft. per min.
Pressure per inch of tooth face.....	37 pounds
Power constant	1.738

Suppose we replaced this by a floating-frame gear with a three-bearing pinion, having the same pitch diameter and speed, the following would be quite safe:

Length of each helix....................	5 inches
Pressure per inch of tooth face...........	326 pounds
Power constant	4
Horsepower	691

That is, we add one pinion bearing and a floating frame; we discard one large gear; about half the tooth face of the pinion and remaining gear; get a stronger tooth by changing the helical angle to 30 degrees; increase the tooth perssure nearly nine times; and more than double the power. All this is accomplished by what Mr. Peterson calls a slight modification of Dr. De Laval's original design and an inexcusable complication. Further, I was reliably informed when I was given the data, that the 300-horsepower De Laval gear had frequently given trouble. I believed that the low values given above were due to a great chance of misalignment occurring even though properly aligned at first; to the great difficulty of avoiding cross-bending due to the powers transmitted by the two gears becoming unequal; and to unavoidable torsion in the long slender pinion. But Mr. Peterson assures us that "No trouble whatsoever has been experienced with uneven distribution of tooth pressure, due to bending or torsional displacement, or to misalignment after the gear reduction has once been properly aligned." I am at a loss then to account for the result. Mr. Peterson surely knows that it is one thing to have experience; a higher, to correctly interpret the experience; and a third, and more difficult thing, to make the proper advance which the experience properly interpreted should suggest.

My reference to the 300-horsepower gear shows that I was aware De Laval gears of over 200 horsepower had been built.

But I still believe that far their greatest success was attained at powers not exceeding 200 brake horsepower. I said it had been confined to small powers. The Westinghouse Machine Company rarely build other than rigid gears for small powers—that is below 500 horsepower—but of course use very low power constants. Therefore Mr. Peterson's statement and mine practically agree, though he does speak of a 750-horsepower gear which was never made! The 1000-horsepower gear which he cites was an ordinary rigid gear, not of the De Laval type at all. No doubt the pinion had two bearings as these alone are figured in the De Laval pamphlets up to 1913; but, as I point out, the three-bearing pinion is much superior unless the helix length is very short. It should be noted also that the date of this 1000-horsepower design was long after that of our gear which was tested to 6000 horsepower.

I have never questioned that rigid gears can be built, or are built in great quantity. They date, as I show, from the third century B. C. I have never limited the power for which they might be built; indeed, I stated "I can see no limits to the horsepower transmissible by a floating-frame gear or rigid gear except those arising from the size of the parts which can be machined in any shop." (*Engineering* (London), v. 103, p. 603.) I nowhere draw "a lurid picture of the evils due to cross-bending, torsion and misalignment" but, if steel and iron are elastic, these evils exist and must produce important effects where teeth are necessarily so rigid. That we now frequently use power constants of 3.5 and 4.0 and have had 100 per cent. success, while rigid-gear builders rarely or never approach these values, is ample proof that the floating frame preserves good contacts always. Mr. Peterson's going back years to find lower power constants

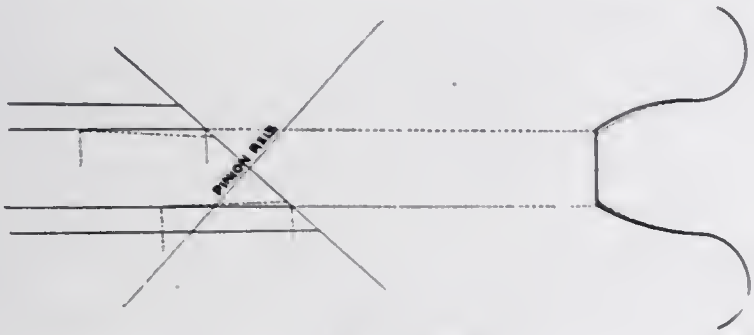


Fig. 23.

proves nothing that is not stated in my paper. If misalignment does not have a detrimental effect in rigid gears why does Sir Charles A. Parsons chamfer the ends of his teeth as shown in Fig. 23?* Did Mr. Haig imagine the failures of gears or the doubts of their lasting qualities? Why do rigid-gear builders use high-class alloy steels? If there is no trouble with rigid gears why is the United States Emergency Fleet Corporation making a careful investigation of the subject? "To obtain the finest results one of the pair of wheels—the smaller one for convenience—should be permitted a little automatic axial movement, and also a rocking movement about a fulcrum in the central plane of the wheel."†

Mr. Peterson cites the tooth pressures on the *Malmanger* and the *Ciudad de Buenos Aires*. He should know that a marine engineer learns very early that there are three powers properly belonging to a ship: 1. The power on the highest measured-mile run. 2. The power for the guaranteed speed. 3. The power usually used at sea.

I have stated these powers in order of magnitude. They usually differ a great deal. For instance, the *Vespasian* reached 1095 horsepower on the measured mile, for a few minutes—power constant = 3.0. Her usual power at sea is 630 with a power constant of 1.85. (See *Engineering* (London), v. 101, p. 492, which also gives data of the *Ciudad de Buenos Aires*.) The *Ciudad de Buenos Aires* has high-pressure pinions of 6.233 inches diameter by about 28 inches total length of both helixes. On the highest measured-mile runs, each of these transmitted 1450 horsepower at 2320 r. p. m. for a few minutes. This gave a power constant of 2.58 and a tooth pressure per inch of 451 pounds. Her service conditions are doubtless much lower than this.

The first-reduction pinions of the *Malmanger* are 5.114 inches diameter by 20 inches total helix length. Her guaranteed continuous horsepower is 1450 per pinion at 3860 r. p. m. This gives a power constant of 2.81 and a tooth pressure of 463 pounds

*I am informed that this was suggested by a British Navy engineer as the ends of the teeth of the gears in his ship were breaking off. Misalignment makes the most intense pressure come at the ends, and extending the tooth beyond the contacts would obviously strengthen it.

†"Toothed Gearing" (chapter on helical gearing, p. 90), by George T. White, B. Sc. (Lond.). Scott, Greenwood & Son, London, 1912.

per inch. The corresponding pressure for a similar pinion of 6.233 inches diameter is 564 pounds; therefore the continuous guaranteed pressure is 25 per cent. above the highest performance of the former ship, and no doubt much more above her service conditions. Thus the floating frame shows to considerable advantage—but this is not all.

Three standard merchant marine designs of gears had been made in July or August 1915, by the Westinghouse Machine Company, for 750, 3400, and 5500 horsepower respectively, with a power constant of 3.3. The machinery of the *Malmanger* was ordered shortly after and it was determined to use standard frame no. 2, partly to save pattern-making and partly because the ship was very heavily boilered for her power, which consequently might be considerable exceeded. Therefore, taking the desgined power of the *Malmanger* gear, the above 25 per cent. becomes $125 \times 3.3/2.81 - 100 = 47$ per cent.

If the service power constant of the *Ciudad de Buenos Aires* is 1.89—in proportion to the values given above for the *Vespasian,* it would be a good deal less—the 47 per cent. becomes 100 per cent. and Mr. Peterson knows from my paper that 3.3 is not a high power constant for a floating-frame gear. Thus this comparison which he adduces seems strongly to support my contention that the floating-frame gear can be safely loaded to from two or three times that at which rigid gears are usually run. The *Malmanger* was an early ship and no special effort was made to save weight or space. Therefore, the assumptions Mr. Peterson made regarding her had no foundation.

As disproving my claim for high and safe tooth pressures and power contacts in our gear, Mr. Peterson refers to certain "municipal pumping jobs". He neglects to mention that the tooth pressures were specified; also certain turbine features patented by the De Laval Steam Turbine Company, so that it is hardly credible that the specifications were not written under their direction or that Mr. Peterson was unaware of the conditions imposed. If this is correct the proposals they submitted should show what their preferred practice is. The following are the figures on

which they were awarded the contracts. I add the tooth pressures and power constants.

TABLE IV

			Pinion					
Place	Date	Power	Pitch-line diameter	Length of face	r. p. m.	Helical angle	Pressure, pounds per inch	Power constant
Cleveland	Feb. 1914	635	5.0″	20″	3600	45°	222	1.41
Cleveland	1914	1065	6.0″	24″	3430	45°	272	1.44
St. Louis	Jan. 1917	1200	6.6″	24″	3600	45°	265	1.16
St. Louis	1917	1800	6.6″	24″	3600	45°	398	1.74

These are very modest pressures and power constants. Further on he refers to a De Laval gear at Ross Pumping Station, and gives, for a 6.6-inch pinion at 600 r. p. m. and 600 horsepower, 955 pounds per inch (3.48 power constant). How this reduction of speed could justify any such rise is a mystery, and I would suggest that Mr. Peterson revise his figures. I have been unable to verify them for this discussion, except that the 600 horsepower seems to be a large overestimate. Mr. Day's 7000-horsepower gear is even more remarkable in this respect, as the figures he gives imply a tooth pressure of 4085 pounds per inch! This is certainly remarkable if the 7000 horsepower was applied over considerable periods. I regret that he has not discussed it more fully.

I strongly differ from Mr. Peterson as to the relative advantage of the small pinion over a larger and shorter one. Especially in the great marine field is a small pinion (that is, a high power constant) valuable—for the same ratio of reduction the gear is lighter and more compact, both of which are valuable qualities on a ship. For similar gears with the same r. p. m. of pinion the weights are inversely as the power constants. Also, for the same size of large gear and r. p. m. of propeller, the turbine revolutions are greater, and consequently the turbine smaller and more efficient, again decreasing dead weight of machinery and fuel, and adding to the earning power of the ship.

I may quote from a letter received recently from Mr. Gerald Stoney, Professor of Mechanical Engineering, Victoria University, Manchester, England. Professor Stoney was for twenty-four years with Messrs. C. A. Parsons and Company and is well acquainted with their turbine and gear practice. He says "As you say, it is the custom to use high-class alloys for the pinions of rigid bearing gears; and if your gear enables mild steel to be used instead, it is obviously a great advantage. The use of small pinions, and therefore of high turbine speeds, is most important."

Mr. Peterson's argument regarding the small increase of weight by substituting a larger and shorter pinion is misleading, but it would take longer to discuss it fully than its importance deserves; for all the data I have seen show that short pinions (that is, much less than two diameters for each helix) are rarely used for such gears as we are considering. The data given in this discussion might be cited. Hence the consensus of opinion is against him, and the table just given proves that the De Laval practice is no exception. I will merely say that if the pinion is shortened and also the power constant lowered, as required by substitution of a rigid gear, the increase of weight will be found to be important. His assumption that he can use the same tooth pressure is incorrect. Possibly manufacturers of rigid gears *could* use the same diameter of pinion as is used in floating-frame gears, but they don't, experience not having bred in them the same optimism which Mr. Day credits to us. I will make definite the increase of weight caused by the floating frame (which he states very indefinitely is "quite an increase"). On a single reduction of 15 to 1 the addition would be about 6 per cent.; in the experimental gear, a 5 to 1 reduction, it was about 17.5 per cent. Increase of volume would not greatly differ from these percentages. On the other hand, halving the power constant, by enlarging to a similar gear with the same power and revolutions, would increase the weight and volume 100 per cent. As confirmatory proof that our gears are smaller I again point to the low power constants and light pressures in table 4, and I may now state that of the three weights offered for a marine installation (noted in my paper, p. 23), built by the Westinghouse Machine Company and weighing 25 000 pounds, the 40 000-pound

offer came from the De Laval Company. Mr. Peterson should know that the weight of a reduction gear of a given power depends very largely on the factors of safety used; on the r. p. m. of the first pinion and total ratio of reduction; and, last, on the general arrangement adopted. Consequently the 32 tons he gives as the weight of the 3000-horsepower (maximum) double-reduction gear means nothing at all on account of necessary data which he has omitted, and the conclusion he so confidently draws from it is unwarranted. Imperfect data also prevent intelligent discussion of Mr. Day's comparison with the weight of the *Malmanger* gear, but happily the same criticism was offered before, where the design was given (See *Engineering* (London), v. 103, p. 333. My reply is given on p. 603). The design was by Mr. Day's old firm in England, The Power Plant Company, Ltd. The power constants are low, but I show that the factor of safety has also been greatly reduced and that lightness and compactness are attained by the spindle, carrying the large gear of the first reduction and the pinions of the second reduction, having no middle bearing, and thus being subject to great cross-bending. To prevent further spread of the bearings carrying this spindle the second-reduction pinions were made narrow, giving a high average tooth pressure, which made a bad combination with the excessive cross-bending. Had these faults been corrected the weight and space would have been much increased. Possibly Mr. Day has made a similar design. Since the gear of the *Malmanger* was not specially designed to save weight, I offered in my reply the gear shown in Fig. 18 and 19. This is of the same general design as the Power Plant Company's but with no sacrifice in any feature. Without the Kingsbury thrust block it weighs 34 tons. As stated in my paper, the reduction is from 3740 to 72 r. p. m. or nearly 52 to 1. The power is 3400. The power constants for this power are 3.28 for each reduction so that saving of weight is not pushed to a high degree. But Mr. Day makes another comparison (p. 53), between a rigid gear and a floating-frame gear. He must have before him the design of a gear in which the centers had to be spread in order to get the high- and low-pressure turbines side by side. It is quite proper that the floating-frame gear should have thinner metal and lighter housing as its success does not

depend on trying to attain perfect rigidity. The bearings may also be smaller since the forces they have to bear are definite. But that is far from all. Mr. Day has chosen a floating-frame gear with a power constant rarely used. If he raises the power 63 to 86 per cent., giving a power constant from 3.5 to 4.0, the comparison will be very different but more nearly correct.

Mr. Peterson objects to my statement that we could have loaded the experimental gear to at least 18 000 horsepower with safety. My statement is based on my belief that the "so-called" power constant is correctly deduced from well established principles, and no erroneous step has been pointed out, and that for such a test as this a value of power constant could be used between 4.0 (not infrequently used and of which I give examples in my paper) and the value 5.22 which has given no trouble on the United States collier *Neptune*. At 6000 horsepower and 1500 r. p. m. of pinion the power constant is 1.48. Consequently at the same speed and 18 000 horsepower it is 4.44 with an average tooth pressure of 2715 pounds per inch. I suspect that all rigid-gear builders have had, at some time, experience of much more than 2700 pounds per inch. As Fig. 3 shows, it is the maximum value, approximately, if the mean pressure is 680 pounds per inch and half the tooth length bearing; or if 450 pounds per inch with one-third of the tooth bearing, both of which assume an exceedingly small misalignment. No one has pointed out where the dynamics of the floating frame is erroneous, vitiating the conclusion that this partial or very uneven contact is completely avoided by its use.

I cannot understand why Mr. Peterson introduced Mr. Rapelye's evidence. Mr. Rapelye was a salesman of the Westinghouse Machine Company who, evidently, did not completely understand the functions of the floating frame. The engineers of the Company were quite clear on the subject. If Admiral Melville or I had ever said or written that bad tooth cutting had anything to do with the subject it would have been good evidence to produce. We alone could have first-hand knowledge of the facts. Mr. Peterson would sympathize with the mental attitude of the prisoner who, when the prosecution produced a witness who saw him commit a theft, offered to produce a dozen who did not see the deed!

I entirely agree with Mr. Day as to the importance of ease of repair. Every machine should be designed with this in mind. There is no such deficiency as he seems to imply, in our gears, but there has so far, I am happy to say, been little occasion for repair.

I am not surprised that Mr. Day found 11 100 feet per minute quite practicable. No indication of a limit was found at the 9600 feet per minute which I give.

With regard to the expansion of the adoption of the floating frame I may say that, long before the recent large orders by the United States Government which have filled up to the limit all who can cut gears, it had so far advanced as to force the purchase of ground for the new works at Essington. It is making friends rapidly and has never lost one. The latest proof of this rapidly growing favor is the license being arranged by the Japanese Government after full investigation of the results obtained.

I will conclude this long reply by a few remarks on the quotations from Sir Charles A. Parsons and Mr. Ferranti in so far as they are not answered in what precedes. Mr. Ferranti, like many others, thought that the floating frame was invented to compensate for imperfect gear cutting. He cannot have studied very carefully the article in *Engineering* (London), v. 88, p. 377, which is no doubt where he got his information, as, while it gives the theory of the floating frame, it contains no reference whatever to bad gear cutting. How he or Sir Charles A. Parsons concluded that difficulties of gear cutting could be completely overcome by the creeping table is remarkable. Its object is to so combine two sets of errors as to get an exact result. Since the combination is fortuitous, while it might reduce the errors (or conceivably increase them), it can never completely eradicate them. Perfect accuracy can be attained only by accuracy of each part of the machine. Those who have seen the gears cut on the creeping table and those cut by the Westinghouse Machine Company have no doubt that the finish of the latter is superior.

Sir Charles A. Parsons judges that he has "fair uniformity" of pressure distribution by examining the teeth. I have shown (*Engineering* (London), v. 102, p. 527) that such an inference cannot be properly drawn. A grinder may polish a large surface uniformly but we cannot therefore conclude that the grindstone

was bearing all over the length of the surface uniformly at one time. Besides, to attain "fair uniformity" of pressure is not satisfactory—the floating frame does much more. Again, everyone who has had to do with gearing knows that what he describes as a "reasonably accurate alignment of the shafts" in a rigid gear must be a process of great precision and it is readily upset. A rigid gear can, of course, be properly aligned and, if it could be so maintained with certainty, it could safely use a high power constant. That the power constant is reduced is proof that there is great disturbance of tooth pressure by slight changes of alignment and even with low power constants trouble is not infrequent, as I could readily prove. Finally, end freedom of the pinion assures, as I showed, equal division of power between the helixes. Sir Charles A. Parsons apparently thinks this is all that is necessary, but surely if one is asked to find how much he can hold up in his hand it should be stated whether the weight is to rest fairly on the palm or to hang from the tip of the little finger.

I do not wish in any way to belittle the great part Sir Charles has played in modern engineering, but his thinking as presented in the quotations given is neither clear nor accurate.

Since the foregoing was set up in type the Japanese Government license referred to above, which was so long under consideration, has been signed, and floating-frame gears will in future be used in the Imperial Japanese Navy.

The following is from a statement made (April 16) by Mr. Francis Hodgkinson, Westinghouse Electric and Manufacturing Co.

"Recent overhauling and examination of the propelling machinery of the U. S. S. *Maui* showed conditions of wear and decomposition due to deterioration of oil used in the system.

At several times leaks had been found in the coolers, which caused admixture of salt water with the oil, the resulting emulsification presumably interfering with lubrication. The oil was found by chemical analysis to contain 4.54 per cent. oleic acid, which condition had caused corrosion of the journals, bearings and gear teeth. Proper provision had not been made by the ship-builder for draining and cleaning the system, which was found to be choked with solid matter, the turbine and propeller thrust bearing cavities having been found almost filled solid. The reduction gears were found in excellent condition, no excessive wear or breakage of the teeth having been experienced at any time.

The difficulties experienced demonstrate the need of using great care in designing the oiling systems for such installations, making provisions for proper draining and cleaning, some changes having been found necessary in this instance in order to accomplish this. The necessity of using proper oil and of constantly watching it for evidences of deterioration is also plainly apparent."

THE FLAT-SLAB SYSTEM—A TOPICAL DISCUSSION

Mr. C. A. P. Turner:* As late as 1906, in a paper discussing reinforced concrete floors, in the *Transactions* of the American Society of Civil Engineers, v. 56, p. 253, Captain Sewell said:

> "No extensive system of reinforced concrete floors can be economically designed, however, without the use of either rolled steel beams or else of reinforced concrete ribs, forming together with a portion of the floor slab in each case what is practically a T-beam."

Of twenty prominent engineers taking part in the discussion of this paper, the writer was the only one who took exception to this statement, and he presented, as appears in the printed discussion, his views of a flat-slab and column construction which could be economically constructed for heavy loads and long spans. This construction has since that date been widely introduced in this and other countries, and various modifications of the precise arrangement shown by the writer in this discussion have been used, but all successful floors of this type necessarily embody the basic principle then disclosed.

The principle proposed was that of imitating in a non-homogeneous slab of concrete and metal the action of a true, homogeneous plate, such as steel, upon separated supports. The distinguishing characteristic of the steel plate is equality of resistance, radially and circularly about the supports and about the panel centers, combined with a fine-grained structure.

Flat plates of reinforced concrete on separated posts were found in the abandoned experiments of the art prior to the Sewell paper. Some of these flat plates were reinforced on the suspension system, as in the Austrian patent to Speer in 1900. Others were reinforced as flat arches—for example, the Payne patent of 1875; Far Rockaway power house; Norcross patent of 1902—while again others were arranged with expanded metal, having some reinforcement in the top of the slab over the supports, and sheets arranged in various ways in the lower part of the slab.

*Civil Engineer, Minneapolis.

But these structures, each and all, lacked the above mentioned characteristics of equality of resistance radially and circularly in the two respective tension zones of the continuous plate, and because they lacked this essential to success, experience prior to the date of the Sewell paper furnished the basis for his assertion.

Beam theory of reinforced concrete was at that date as fully understood as to-day; yet, strange to say, the Joint Committee of the American Society of Civil Engineers tries to account for a result—known to be impossible in 1906 on beam theory—by nothing more scientific than this same old beam theory, in which no advance has been made since that date by this Committee or any one else.

Equal column spacing will be considered for simplicity of discussion since this phase of the problem is sufficient to clear up the principles involved.

In beam action, linear resistance to bending is accompanied by cylindrical curvature. Plate action, or imitation of plate action, presents radial and circular resistance combined—i. e., double curvature. This kind of resistance is afforded by a circular plate supported around its edges. The flat slab on column supports when scientifically reinforced may furnish similar resistance, because the bending of the plate so reinforced in a circular area about the column in diameter equal to approximately half the column spacing, will be convex upward, and a similar area about the panel center will be dish shaped, or concave upward, with saddle-back areas between. The change in curvature from the saddle-back portion to the dish-shaped portion involves twisting in the material of the plate.

Thus plate action differs from beam action, as follows:

1. Double curvature under bending for the plate, as against cylindrical curvature in the beam.

2. By presenting circular and radial dual resistance in the plate, instead of linear resistance as in beam action.

3. It involves twisting moments and bending moments combined, instead of bending moments alone as in beam action.

4. By providing storage of energy of internal work in a manner not directly involved in deflection, by which substantially

four times the stiffness and strength of beam action is secured, disregarding Poisson effects, in a square panel in respect to the dish-shaped areas about the column and the diagonal center of the panel. (See "Concrete-Steel Construction," by H. T. Eddy and C. A. P. Turner, v. 1, 1914, pp. 136-141.)

5. Because of this different mode of operation, a wide difference is found in the percentage of external work stored as internal work of deformation, in the steel in slab and beam structures respectively, as all experiments show. For beams, when the maximum steel stress is 16 000 pounds per square inch, this percentage is approximately 60, while in slabs it is less than 12.

6. Figured by the erroneous adaptation of beam theory recommended by the Joint Committee the moment of resistance of the steel is found by calculation to be five to seven times as great as extensometer tests show it to be.

7. On beam theory, restraint at the support (fixed beam) increases the stiffness five times and the strength one and one-half to two times, while a plate panel simply supported at the corners appears from experiments to be but one-eighth as strong and one-twentieth as stiff as though plate continuity and restraint were present.

8. In a beam, the direct tensile stress in the concrete plays such an important role that, with the yielding of the concrete, the deflections present a wide departure from the proportionality of load and deflection as the loads are increased in developing even working stresses in the steel. In the column supported slab scientifically reinforced, because the direct tensile strength of the concrete plays a very unimportant part, there is relatively close agreement between the magnitude of the load and the deflection.

Such radically different phenomena as those just enumerated would seemingly bar the theorist, as it does the practical constructor, from endeavoring to apply mere beam theory to explain such divergent results. The idea, however, that the only reliable tensile resistance possible in the slab must reside in the steel alone, as is the fact in beams, causes many not fully conversant with reinforced concrete to dodge the issue of the joint action and co-operation of the two materials, and endeavor to attribute

all the essential advantages above noted to the tensile strength of the plain concrete slab, combined with mere beam action of the steel.

The paper of H. T. Eddy, *Proceedings* of the American Concrete Institute, February 1916, v. 12, p. 281, demonstrates the well-known law of moment magnitudes for the beam, i. e., half the sum of the moments over the supports plus that at mid-span equals $\frac{1}{8}$WL for uniform loads, and shows that in all published flat-slab tests, taking a panel-wide strip between column lines, the moments of resistance of the steel, figured from measured stresses, aggregate only one-fifth to one-seventh this known fixed quantity present in mere beam action.

In beam tests, United States Bureau of Standards, Technologic Paper no. 2, extensometer measurements were made on the concrete along the plane of the steel. The steel stresses so determined were the average for the length between cracks, and represented 85 to 90 per cent. of the moment resistance. Therefore if the concrete may afford in the slab five to six times the resistance that the steel furnishes, whereas in the beam it furnishes but 10 or 15 per cent. of the resistance, plate action of plain concrete must be 30 to 40 times as efficient as beam action of plain concrete. The supposition that such a result is possible requires a different explanation from that advanced by the Joint Committee or afforded by the traditional flat-plate theory applied to concrete. By theory the plate action of concrete is but 4.04 times as efficient as beam action for $K=0.1$, and 4.16 times for $K=0.2$. Hence the explanation put forward by the Joint Committee is an absurdity. As in beam action there is a law of moment magnitudes, so also in the slab action under discussion there is a similar law, differing from that of a beam in that the internal resisting moments are divided between bending moments and twisting effects. Internal moments are by-products of shears, and shear cannot produce in a slab the same amount of bending moment as in a beam—where there is no twisting—and produce at the same time twisting moments in addition to bending effects of the same amount as in the beam. This proposition should be self evident from the principles of statics and the law of conservation of energy.

The error in reasoning, upon which is based the commonly accepted statement that the bending moment parallel to each side is equal to ⅛WL, is due to the assumption that the moment of the external forces about an axis parallel to a side is the product of ½W (the supporting reaction at one side) by ½L (the arm about mid-span), minus ½W (the load on the half panel) times ¼L (its arm about mid-span). Now this is undoubtedly the statical moment of these forces about mid-span but this statical moment is the sum of a bending moment and a twisting moment so that the bending moment is only a part of this total moment. The vertical reaction at a column (equal to ¼W) is made up of a vertical shear (equal to ⅛W) on a plane parallel to one side of the panel, and a second shear (equal to ⅛W) on a plane at right angles thereto. The first of these two shears produces a bending about an axis parallel to the plane of shear, together with a twisting moment about an axis at right angles thereto. The second shear acts in the same way, about axes at right angles to those just mentioned. We have therefore a bending moment and a twisting moment about each axis parallel to a side, each of which is less than ⅛WL because that is the sum of the two.

Bending moments produce flange stresses or horizontal stresses along the surface of the plate resisted by the combined action of the steel and concrete. Twisting moments, on the other hand, produce torsional shearing stresses. Thus, bending moments and twisting moments follows such different laws that any attempt to combine them, as in effect is undertaken by Mr. Nichols in a paper on the "Statical Limitations upon the Steel Requirement in Reinforced Concrete Flat Slab Floors,"* is wholly erroneous. The treatment of torsional shear by St. Venant may be of interest. This is given in W. J. Ibbetson's "Elementary Treatise on the Mathematical Theory of Perfectly Elastic Solids." 1887. MacMillan.

The joint action of bending and twisting moments in a homogeneous plate has been treated mathematically by Thompson and Tait, in their "Natural Philosophy," published fifty years ago (1867), pp. 489-491. These eminent physicists have clearly ex-

*Transactions, American Society of Civil Engineers, v. 77, p. 1670.

plained not only the application of the principles of statics to this problem, but the fundamental methods of attack by the theory of work as well.

The Joint Committee report* shows in a clear light the lack of thorough training in our engineering schools. It indicates that fundamental principles for the correct application of the laws of statics and work, clearly understood and taught by the leading physicists of fifty years ago, are unknown, unheard of, and untaught in the present-day schools of engineering of the universities of Illinois, Purdue and Wisconsin. A broader curriculum, embracing a thorough course in the principles of elementary physics and mechanics, would render the principal phases of the mechanical action of flat plates intelligible, instead of presenting the enigma they do to-day to the confusion of engineering professors and builders; by reason of the test results shown by a flat-slab floor in Chicago as noted in *Popular Mechanics,* November, 1917, p. 657. The pitiable performance of a well-known engineering research fellow of a prominent university—testifying as an expert for the enlightenment of the court—who expressed his disbelief in any possible relation between the internal work of deformation and the external work of the load in a flat-slab floor, because he knew of no way to measure this relation with an extensometer, gives point to the above suggestion.

Having determined the total magnitude of the applied forces it should be noted at once that the maximum applied moment across the panel edges cannot be materially greater with ordinary arrangements of reinforcement than 1/24 WL, while that at mid-

*The writer has reached an exceptionally clear understanding of the unscientific basis of the Joint Committee report on the operation of flat slabs in the light of the testimony (founded on a total disregard of the law of least work and the principle of rigidities) of a member of that Committee, in the patent suit of Drum vs. Turner, to the effect that the mode of operation of a column-supported reinforced flat slab would be identically the same, before the concrete cracks, whether the steel were disposed in the top of the floor or in the bottom thereof. Accepting this testimony as being in accordance with the most advanced engineering opinion, the Eighth Circuit Court of Appeals, with flawless legal logic unhampered by even a trace of engineering understanding, held (Federal Reporter, v. 219, p. 192, Eighth Circuit Court of Appeals, 1914) that disposing the steel in the top of a floor was the plain mechanical equivalent of placing steel in the bottom of a floor, thus unerringly following the judical precedent, established as late as the seventeenth century, of reversing those mechanical principles taught by the illustrious philosopher Galileo, who, it appears, is to be accredited as the first to give a clear exposition of the operation of stresses in the flexure of beams and the like. (See Encyclopaedia Britannica, "Galileo.")

span cannot be more than 1/48 WL, providing there is fixity or proper restraint at the supports. The condition of restraint, however, is to be investigated as an important practical problem, for if it is lacking, the moment at mid-span increases under unbalanced loading.

The moments for unbalanced loads must be apportioned between the columns and the slabs in accordance with the law of rigidities. Thus their proper distribution can be mathematically determined when the column sections and the moment of inertia of the slab are known. This is a most important investigation because experimental tests show, as theory also indicates, that with the same slab and the same reinforcement nearly 100 per cent. of variation may occur in the deflection under a given load applied to a single panel only, where there is a wide variation in the rigidity of the supporting columns.

The latest research work of Dr. H. T. Eddy has solved this problem from the mathematical standpoint in a very satisfactory manner indeed, the writer's prior efforts at solution having been based on approximate proportional methods which would not bear too close scrutiny from the mathematical standpoint, notwithstanding that he has successfully built column supported slabs of from 15- to 50-foot span.

The principle of imitation of plate action is that of the operation of a framework or mat of small crossed rods of proper area embedded in a concrete matrix in the two tensile zones, the rods being of such small size and close spacing as to approximate to a fine grained homogeneity of structure, on the same principle for example as that by which a fine screen enables the making of a printer's cut which will reproduce the light and shade of a photograph. Yet when this print is examined under the microscope we see nothing but a series of dots. So the general effect of a homogeneous structure is secured by that particular arrangement of metal above outlined. Uniformity of the up and down dishing must be secured by uniformity of resistance radially and circularly; otherwise no greater efficiency can be secured than may be accounted for by mere beam action. In the fact that in early attempts to build flat-slab floors, these necessary requirements

were neither understood nor complied with, is found the justification for the assertion of Captain Sewell in the year 1906.

We have stated that our framework or mat of steel furnishes resistance through the operation of radial and circular stresses made possible by the combined action of the metal and concrete. The concrete through bond shear resistance so combines the elements of the mat that circumferential deformation is resisted jointly by tension in the steel and by bond shear resistance between the steel and the concrete, closing the polygons, as it were. The forces of resistance in the tensile zone are thus divided equally between tension in the steel elements of the mat, and bond shear resistance which prevents circumferential separation of the elements of the mat, and since in flexure the horizontal resistance in the tension and compressive zones must balance the concrete in the compressive zone, it is required not only that the compressive resistance balance the tension in the steel, but also the bond shear resistance which operates to close the circumferential polygons formed by the elements of the mat of crossed rods; and the steel in conjunction with the matrix under these conditions then acts essentially as though it were a sheet of steel. Now the operation of a sheet presents this characteristic, that it can resist, at one and the same time, the same amount of tension in each of two directions at right angles to each other, that it is able to resist when this tension is applied in a single direction alone—and in fact through Poisson action it resists the two forces at right angles with greater efficiency than it can resist one alone acting in either direction. Therefore the continuous plate of reinforced concrete requires only half the steel, to develop the compressive strength of concrete, that it required on beam theory, but this fundamental principle is ignored by most members of the engineering profession, and by building codes, resulting in great waste of steel in structures of this kind, and frequently in dangerous designs embodying an unscientific, unbalanced relation of the steel and concrete elements.

In conclusion, the only solution of the problem of the flat slab, which is in substantial agreement with experimental results, is that developed by the joint work of Dr. Eddy and the writer, in which the writer has treated the matter from the standpoint

of the theory of work, while Dr. Eddy has treated it by an adaptation of the traditional flat-plate theory. This method involves what Mr. Godfrey of Pittsburgh has been pleased to characterize as the "mysteries of Poisson's ratio." In the homogeneous material, Poisson effect is directly related to the co-efficient F, of shearing rigidity, and in the composite material of concrete and metal, F, in the tensile zone, must be calculated upon the combined resistance of the steel and the concrete there present. The steel represents about 1/50 of the cross-sectional area of the concrete in a given direction, and as co-action with the concrete through bond causes little cup-shaped deformations close to the surface of the rod, and very much smaller deformation between the rods, the effect upon deflection is that of the average deformation which is 1/12 to 1/15 that of the maximum next to the surface of the bar. Consequently the shearing rigidity of the concrete plate as a whole, with respect to the steel, is 10 or 15 times greater than that of the shearing rigidity, F, of the plain concrete in contact with the steel, and the Poisson effect to be taken into consideration in computing the deflection of the composite structure must be that based on the mean value of F for the combined materials, and not upon that for steel or for concrete as separate elements.

Attempts to determine the magnitude of F experimentally where there is cylindrical curvature as in a beam, will of course result in failure, for should any one discover such Poisson action in beam curvature it would upset entirely the basis of the theory above outlined. Professor Slater of the University of. Illinois, made experiments on a small model which he called a control slab, supported like a beam on two ends, bending in cylindrical curvature, and finding no combined Poisson effect such as is found where there is double curvature, he assumed that he had disproved the Eddy-Turner theory—instead of discovering in this, indirect evidence tending to substantiate the theory. For those who find difficulty in accepting the above explanation of Poisson effect, which it is necessary to consider to compute deflections with precision, it may be suggested that disregarding K entirely, or treating its value as zero, results in only a moderate reduction (25 per cent.) in the efficiency of the plate as compared with the

beam—so that the mysteries of Poisson's ratio need not prejudice any against the use of an advantageous and economic type of flooring.

Imitation of plate action differs from true plate action in the zones where the lateral efficiency due to this Poisson effect is most marked. Bond shear is zero at the center of the span and over the supports in the continuous beam, because the moment at these points passes through a maximum, but is greatest at and near the point of inflection. Similarly the greatest interaction of stresses at right angles occurs at the outer edge of the cantilever and the end of the suspended span portion, or in zones about the one of inflection. This is the basis of the marked difference which Mr. Smulski found in the apparent efficiency of his spiral coils in the outer portion of the cantilever and in a mat of rods which did not extend circumferentially about the outer portion. He had in fact experimentally discovered the principle, the meaning of which he did not understand. His experiments were published in part in the *Engineering Record* as the writer recollects it, about two years ago.*

The writer has extended his paper beyond the limitation originally proposed, because the flat slab is a subject upon which a volume might readily be written, inasmuch as years of study have been expended in its development. An effort has been made to present a brief survey of the subject and to explain in a simple manner the characteristics of plate action not generally understood.

Mr. E. S. Martin:† The development of reinforced concrete construction in America as well as in Europe has been largely through firms and individuals exploiting proprietary so-called "systems." In one respect this method of development has been most unfortunate. Technical discussion of the different forms and details of concrete construction has been most unfairly affected by professional jealousies as well as commercial antag-

*Engineering Record, Feb. 12, 1916, v. 73, pp. 217, 247.
†Resident Manager, C. A. P. Turner Company, Minneapolis.

onisms. It is within three years that a scathing criticism appeared in an engineering journal directed at the ruling on flat-slab design in use by the building department of a large city, this criticism being based on the mere assumption that the ruling allowed as safe a building in another city which was found to be quite unsatisfactory. The point is that the critic was intensely antagonistic toward both the building department referred to and the engineer who had designed the unsatisfactory building; there was in reality absolutely no connection between the two and no basis in fact for the criticism. The question of the proper method of design for flat-slab floors in particular has been more than commonly afflicted by this distemper. The writer believes that this high feeling will quickly subside with the settlement of the patent question within the next year or so.

Furthermore, there is little room left for fierce discussion in view of the large number of carefully conducted tests on completed buildings. These have already established beyond question what the actual stresses are in interior panels, although there are yet unsettled minor questions such as stresses in wall panels with varying spans and sizes of pilasters, bending stresses in columns with unbalanced floor loads, and the effect of changing the position and distribution of slab reinforcement. The writer desires to confine his remarks to the question of the safe design of the slabs.

Most of the building tests have been conducted by the Engineering Experiment Station of the University of Illinois, and reported in its bulletins. Others have been reported in the engineering press and in society transactions. These have shown that the greatest intensity of slab bending occurs at the edge of the column-capitals. The early flat-slab buildings were not provided with the depressed panel and were generally weak at the capital, concrete stresses running from 1000 to as high as 2000 pounds per square inch, and the steel stress at the capital edge from 16 000 to 25 000 pounds under the designed dead and live load. The depressed panel has generally remedied this weakness.

The stresses are highest at the capital edge and about uniform around its circumference except for the embedment of the various layers or belts of rods. Steel stresses are not even approximately

uniform across the belts at the capital; the inner rods of a belt which pass over the capital get the critical stresses while the outermost rods are much less stressed—hardly at all in some cases. The increase of stress in a slab rod as the capital is approached is at a greater rate than occurs in a continuous beam, and the same is true of the concrete compression. The writer pictures the conditions to his mind as a bending-moment variation of the continuous beam and an effective resisting section decreasing as the column is approached. This makes the stresses vary as the cubes of the distances from the capital edge instead of in the ratio of the squares as for a beam. Whether this is entirely correct is not proven but it is at least nearer the facts than is the beam principle.

It is for this reason that the writer considers the Pittsburgh ruling on flat slabs, which has been in force for the past few years, to be much superior to the analysis by belts or strips as in the Chicago method or Joint Committee recommendation. Furthermore, the Pittsburgh method is more elastic, in that no fixed ratios for capital diameter and depression breadth are required, the method itself controlling the safe sizes.

The actual stresses in a typical panel having been determined by the tests, controversy can remain only over the margin to be allowed in designing formulas to cover unreliable aids that may be present in the structure tested—such as tension in the concrete, arch action and the like. It would seem fair that the same allowances should be made for flat-slab design as are provided by the standard accepted method of designing beam-and-slab floors. In Tables I and II are given results of tests on simple beams. It is seen that in the beam of Table II, the effect of concrete tension, etc., died out when the measured steel stress much exceeded the standard safe working stress of 16 000 pounds. This is not true for the United States Bureau of Standards beams in Table I. Here the computed stresses are from 25 to 50 per cent. above the measured values, even when the measured stresses are high. Also the percentage of steel has little effect on this ratio except for extraordinarily high percentage of reinforcement.

Again the ratio of tensile steel to tensile concrete (entire slab), at the supports of a beam or girder in a building, is quite

TABLE I

U. S. BUREAU OF STANDARDS—TECHNOLOGIC PAPER 2

	Beam 166		Beam 167	Beam 178		Beam 179	Beam 189		Beam 190	Beam 201		Beam 202
	% .49		% .49	% .74		% .74	% .98		% .98	% 1.30		% 1.30
Load	Meas.	Comp.	Meas.	Meas.	Comp.	Meas	Meas.	Comp.	Meas.	Meas.	Comp.	Meas.
1100	1020	6800	1080	1140	4640	1290	1080	3550	630	960	3070	1170
2000	1800	13600	1860	1860	9280	2010	1740	7100	1290	1650	6140	1950
3000	2550	20500	2760	2670	13950	2880	2460	10700	2220	2310	9200	2760
4000	4130	27300	7150	3840	18600	3700	3400	14200	3180	3030	12300	3900
5000	20200	34000	22200	6860	23200	5350	4980	17800	4920	4560	15350	5700
6000	30800	40800	30400	14900	27900	11100	8460	21300	8750	7500	18400	8700
7000				20800	32500	17650	13800	24900	14300	12200	21500	12900
8000				26000	37100	23300	17900	28400	19700	16300	24600	17000
9000				31400	41800	28500	21800	32000	24400	20500	27600	20600
10000				37600	46400	33000	25800	35500	29400	24100	30700	24500
11000							29700	39100	34000	27500	33800	28700
12000							33800	42600	38300	31400	36800	32500
13000										34900	39900	36500

	Beam 213 % 1.6		Beam 215 % 1.6	Beam 225 % 1.84		Beam 226 % 1.84	Beam 243 % 2.12		Beam 245 % 2.12
Load	Meas.	Comp.	Meas.	Meas.	Comp.	Meas.	Meas.	Comp.	Meas.
1000	1170	2630	1290	930	2240	1020	1140	2010	1290
2000	1860	5250	1860	1650	4500	1650	1590	4020	1890
3000	2610	7800	2610	2310	6730	2400	2250	6020	2550
4000	3690	10500	3600	3240	8960	3030	3030	8030	3240
5000	5130	13150	5220	4410	11200	4200	4050	10100	4200
6000	7230	15800	8300	6150	13450	5700	5370	12100	5430
7000	10000	18400	11400	8660	15700	7900	7160	14100	7320
8000	13200	21100	14800	11390	17900	10700	9180	16100	9120
9000	16500	23700	17900	14380	20200	13950	11400	18100	11650
10000	20000	26300	21300	17350	22400	17250	14200	20100	14300
11000	22500	28900	24700	20500	24700	19900	16700	22100	16900
12000	25700	31600	28000	23200	26900	22600	19300	24100	19500
13000	28900	34200	31100	26000	29200	25200	21800	26100	21900
14000	31800	36800	34300	29000	31400	28000	24200	28100	24100
15000	34700	39400	37500	31600	33600	30700	26600	30100	26400
16000	38300	42100	41000	34600	35900	33300	29000	32100	28700
17000				37100	38100	36200	31200	34100	31000
18000				40600	40300	39700	33500	36100	33400
19000							36400	38200	35900
20000							40200	40200	39100
21000							46900	42200	43200
22000									51600

TABLE II

UNIVERSITY OF ILLINOIS ENGINEERING EXPERIMENT STATION

BULLETIN 28. OCT. 5, 1908, P. 16

Beam 72 Percentage of steel 1.27

Measured.	Computed.
1200	6300
2100	8000
3600	9700
6000	11500
7500	13400
10500	15300
14700	17400
17700	19300
20100	21400
23100	23300
25800	25300
27900	27300
29400	29300
31800	31400
36900	35300
42000	39300
47700	42900
61200	47600
108000	49600

low and the effect of tension in the concrete accordingly great. For the flat slab this is reversed, because the concrete section is a minimum around the periphery of the capital and the reinforcement is here concentrated, the percentage of reinforcement being from 0.6 to 1 per cent.

From this study the writer believes that an allowance in the bending moment coefficient for flat slabs at the capital, giving computed values 10 to 25 per cent. above measured stresses where these are high, is safe and fair; and a margin of 50 to 100 per cent. for the middle of the span, likewise adequate. It is not

claimed that these allowances are sufficient over the measured stresses in the cases where the latter are low, say under 10 000 pounds, as in such cases no conclusions should be drawn, the tensile effect of the concrete being indeterminate.

Mr. T. L. Condron:* In December 1912 the writer read a paper† at the Pittsburgh meeting of the National Association of Cement Users, now the American Concrete Institute, entitled "Principles of Design and Results of Tests on Girderless Floor Construction of Reinforced Concrete." In that paper, which was abstrated in *Engineering and Contracting*, January 1, 1913, the accompanying Fig. 1 was used as an illustration. The writer believes this to be the first published suggetsion of an analysis for stresses in flat slabs in which the panel was subdivided into rectilinear strips. The writer at that time said:

"Fig. 1a is a diagram of a portion of such a floor supported upon six columns and Fig. 1b is a longitudinal section across the same, while Fig. 1c is a diagram of certain moment curves. If a longitudinal element of the floor, in line with the column centers, be considered as uniformly loaded and the width of the column heads be reduced to knife edges, then the bending moments along this element of the floor will be represented by the dotted curve, in Fig. 1c, and the negative moment over each support will be equal to WL/12, where L is the distance between knife-edge supports. At the center of the span, the positive moment will be WL/24, or one-half the moment over the support. If the supports are wide column heads instead of knife edges, the moment curve will be changed, because the clear span will be shortened and this element of the floor will become a beam with fixed ends and a span equal the clear distance between column caps. The moment at the edge of the column cap will be equal to WS/12 where S equals the clear span, and the moment at the center of the span will be WS/24, or one-half the moment at the edge of the cap . . . If the columns are placed in the usual manner, as indicated in Fig. 1a, each column will support a section of floor, C_1, C_2, C_3, C_4; and if the column heads are rectangular, there will be supported on each edge of the column head an area of the floor equal to A, C, C_1, A_1.

*President, Condron Company, Chicago.

†Proceedings, N. A. C. U., v. 9, p. 116.

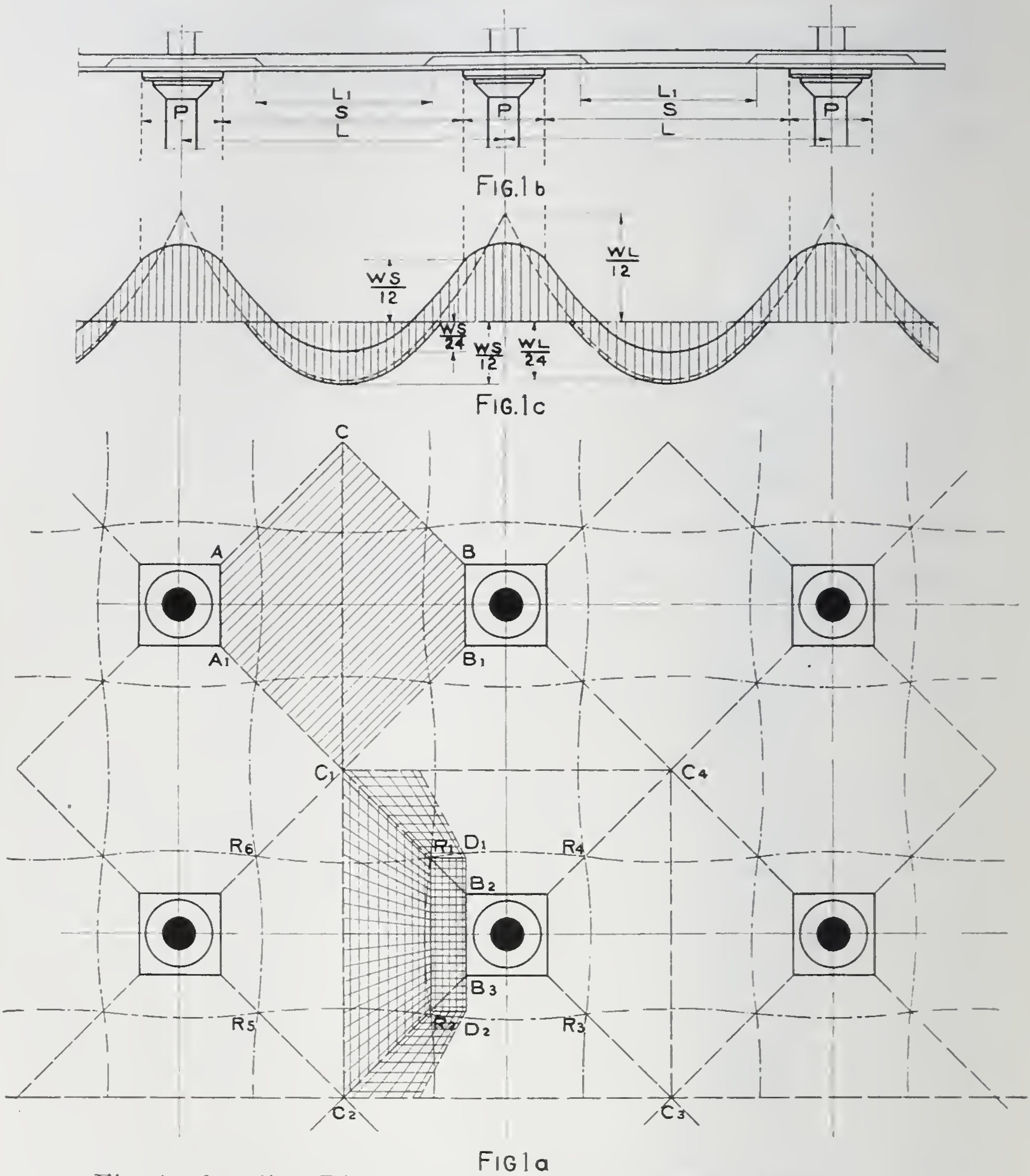


Fig. 1. Loading Diagrams and Moment Curves—Used in 1912.

"Two simple ways are suggested for analyzing the stresses in such a slab supported upon rectangular column heads and with rectangular reinforcement, either of which follows the ordinary method of beam analysis, does not involve abstruse reasoning, and may be relied upon to give safe and sane results, nearly enough correct for all practical considerations.

"The first method is to consider the loaded area A C B B_1 C_1 A_1, carried by the broad girder A A_1 B_1 B having the clear span A B and having its ends fixed. Considering this girder uniformly loaded, the moment at the edge of the column plate will be WS/12, since W equals the load on one-half the panel (exclusive of the area of the column plate)

it is equal to $W/2\ (L^2 - P^2)$. Therefore the moment at the edge of the cap is equal to $W/24\ (L^2 - P^2)S$, or $W/24\ (L^2 - P^2)\ (L - P)$. If $P = L/4$, the moment becomes equal to $WL^3/34$. . . . The reinforcement of the region enclosed by the four main belts is readily determined, this portion being a slab reinforced in two directions. The moment between the lines of inflection will be $Wl^2/8$ or $Wl^2/16$ for each direction where l is the distance between lines of inflection.

"The importance of transversely reinforcing the top of the slab, across the main belts of bars, has already been mentioned and too much emphasis cannot be laid upon this feature.

"The proper width of the belts of reinforcement is a matter of judgment based upon experience and deflections under tests, but these belts will be wider than the caps, but not wider than the distance between the lines of inflection."

It is perhaps interesting to compare this early suggestion of a method of determining stresses in flat-slab construction with the most approved practice of to-day, because the entire theoretical development of this type of construction is covered by the past five years. The earliest application of flat-slab construction was about sixteen years ago, at which time some rather crude attempts were made to build floors without the use of girders, and a patent was issued to Mr. O. W. Norcross who apparently was the pioneer. His work was followed by the work of Mr. C. A. P. Turner who appreciated the necessity of placing reinforcement near the upper surface of the slab over the column supports. But, similarly to the Norcross method, he placed his reinforcement in diagonal lines as well as directly from column to column. Mr. Turner did not offer any very clear or simple analysis of stresses and there was no established method for determining these stresses until subsequent to the paper above referred to. A method of analysis was proposed by Mr. A. B. MacMillan in 1910, and in 1913 (May 21) Mr. J. R. Nichols presented his paper on this subject before the American Society of Civil Engineers.* He, however, did not offer any suggestions as to the distribution of total negative and total positive moments in the different portions of the slab. But in 1914 Mr. J. Norman Jensen, Engineer of the Chicago Department of Buildings, published in the *Engineering News*, v. 72, p. 632, the flat-slab ruling prepared under his direction and adopted by the Chicago Depart-

*Transactions, A. S. C. E., 1914, v. 77, p. 1670.

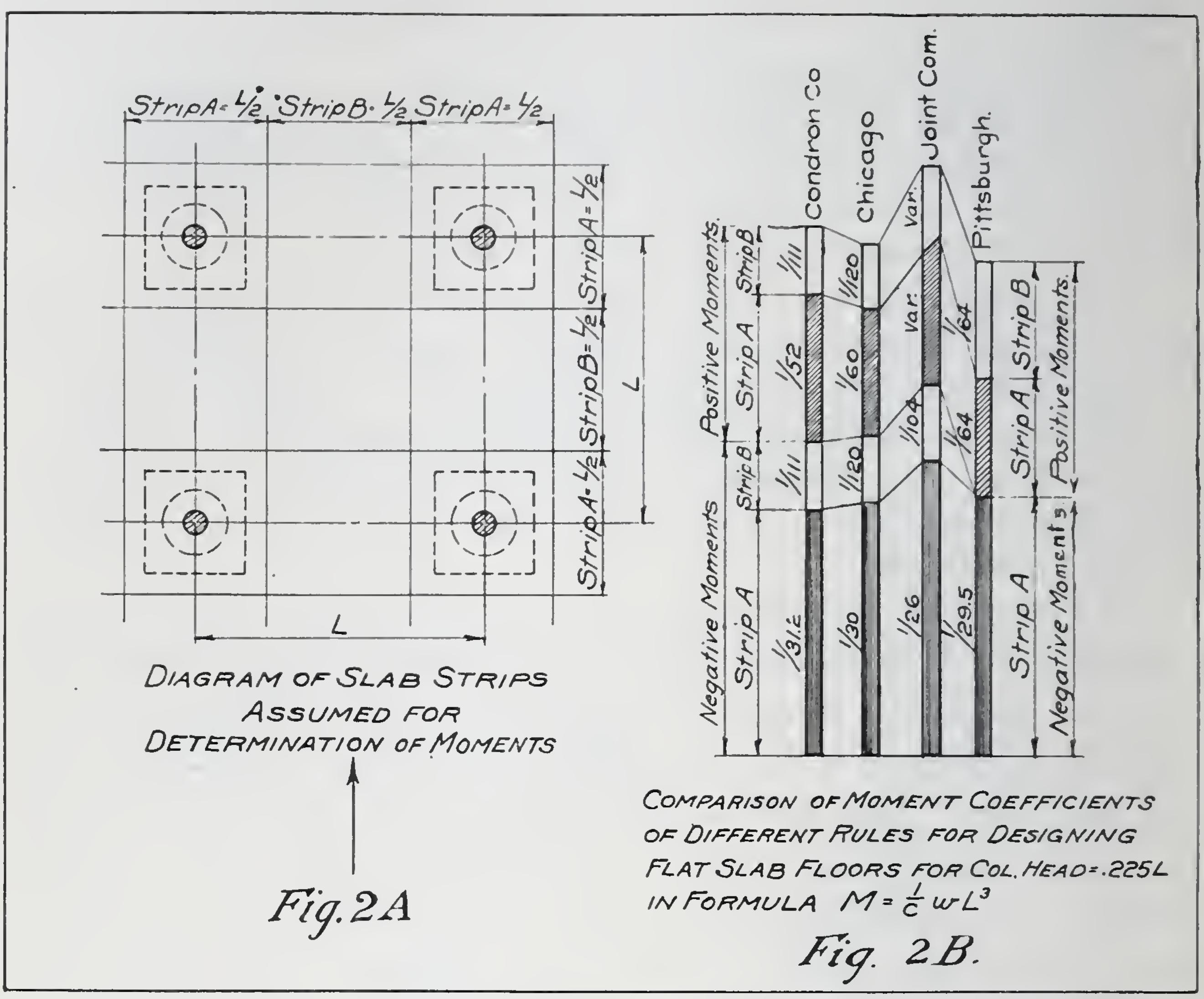


Fig. 2A and 2B. Comparison of Moment Coefficients—Various Rules.

ment of Buildings. This ruling has been very widely followed throughout the United States, and in this ruling the slab was considered divided in to strips A and B (Fig. 2A), in which strip A included the reinforcement and the slab in a width extending from the center line of the columns for a distance each side of this center line equal to one-quarter of the panel length. Strip B included the reinforcement and slab in the remaining width in the center of the panel. At right angles to these strips the panel was divided into similar strips A and B having the same width and relation to the center line of columns as the above strips. These strips were assumed for designing purposes only and not intended as boundary lines of any bands of steel used. In Fig. 2B the writer has plotted to scale the moment coefficients as proposed in his paper of December 1912, with the single modification that the positive moment in strip A is taken as 60 per cent. instead of 100 per cent. of the negative moment in that strip. It also includes

the moment coefficients of the Chicago ruling and those that he understands are required by the Pittsburgh Building Department, and the Joint Committee. The Pittsburgh coefficients have been taken from the article published in the *Engineering News-Record* of December 20, 1917, v. 79, p. 1155, by Mr. John B. Krippner, Engineer, Department of Buildings, Chicago.

It has been the practice in our office to determine the negative moment in strips A by using the formula WS/12 where W is one-half the total live and dead load on the panel, exclusive of the load directly over the column head, and where S is equal to the clear span between column heads, assuming an equivalent square column head where round column heads were used. This explains why the coefficient on the diagram for this moment is shown to be 1/31.2 instead of 1/12; the coefficient, 1/31.2, for WL^3 being equivalent to 1/12 for WS when a round column head having a diameter equal to 0.225L is used.

It will be seen that our moment coefficients differ but little from the moment coefficients required by the Chicago ruling. It will also be noted that the Pittsburgh rules make no provision for negative moment in the strip B and apparently provide an excessive amount of positive moment for the same strip. At present it is very generally realized that provision must be made for negative moment in strips B as well as in strips A.

TABLE III

CONDENSED STATEMENT OF DEFLECTION TEST RECORDS

DESCRIPTION OF TESTS	DEFLECTIONS Maximum	Minimum	Average
12 Tests of 1 Panel each with 2 xL.L.	.0007	.0002	.0005
4 " " 2 " " " 2 xL.L.	.0011	.0003	.0006
3 " " 4 " " " 2 xL.L.	.0007	.0002	.0005
7 " " 2 " " " 2½xL.L.	.0016	.0006	.0011
1 " " 4 " " " 2½xL.L.	.0014	.0010	.0013
3 " " 1 " " " 2¾xL.L.	.0012	.0004	.0008
2 " " 2 " " " 2¾xL.L.	.0014	.0002	.0009
1 " " 1 " " " 3 xL.L.	.0008	.0008	.0008
2 " " 2 " " " 3 xL.L.	.0008	.0004	.0006

NOTE—The above tests include 4 corner panels, 15 wall panels and 45 interior panels, with an average deflection=0.0007 of diagonal span, or 0.001 of average L.

We have the records of a large number of tests on floors built on our designs and Table III is a condensed statement of the deflections measured at centers of panels under these test loads.

It will be noted that these tests have in some cases been made on single panels and in others on two panels, fully loaded, while quite a number have been made on groups of four panels, fully loaded, at the same time. Also that these tests have been made with 2, 2½, 2¾, and 3 times the live load for which the panels were designed. The deflections given are expressed in the ratio of the deflection to the diagonal distance between supporting columns, and the maximum, minimum and average deflections so expressed are given for each group of tests. The average deflection for the 63 panels recorded is equal to .0007 of the diagonal span, or .001 of the average direct distance between column centers. We have the records of tests on 15 wall panels and of four corner panels.

It has been our practice to design wall panels with moment coefficients 20 per cent. greater than for corresponding span of interior panels—this increase applying to the negative moment over the first interior column from the wall, and to the positive moment between the first interior column and the wall column.

In determining the amount of steel required to resist the negative and positive moments we have used the ordinary straight-line formula, and have used such unit stresses as were required by ordinance, in some cases using 16 000 pounds per square inch, but in the majority of cases a unit stress of 18 000 pounds per square inch has been used for steel having an elastic limit of not less than 50 000 pounds per square inch. So far as deflection tests are concerned, we have been unable to discover any difference in the deflection whether a unit stress of 16 000 pounds or 18 000 pounds per square inch has been used in the design.

There has been but one test made of our type of construction in which the stresses in the steel have been measured. This test was made by the Seattle Building Department on the eighth floor of the nine-story building erected for Sears, Roebuck & Company, at Seattle. This test included two wall panels and two interior panels, as shown in Fig. 3. Fig. 4 shows the maximum

Fig. 3. Test of Four Panels—Sears, Roebuck Building, Seattle.

tension and compression stresses under approximately 2½ times the live load. This floor was designed for 200 pounds per square foot live load, and a finished wood floor weighing 25 pounds per square foot, and tested with a test load equal to 550 pounds per square foot. The maximum tension recorded in the main re-

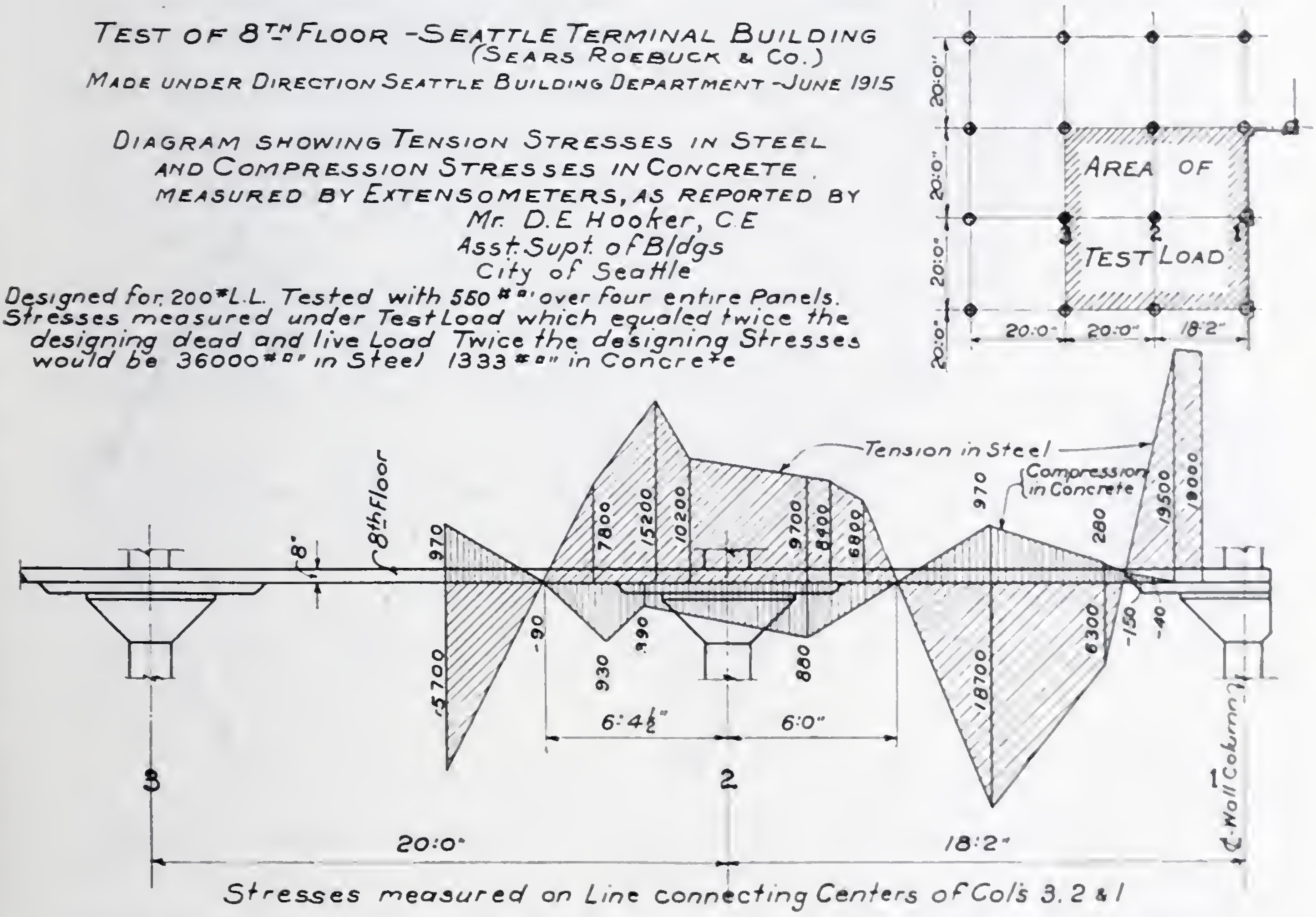


Fig. 4. Diagram showing Maximum Tension and Compression Stresses—Seattle Terminal Building.

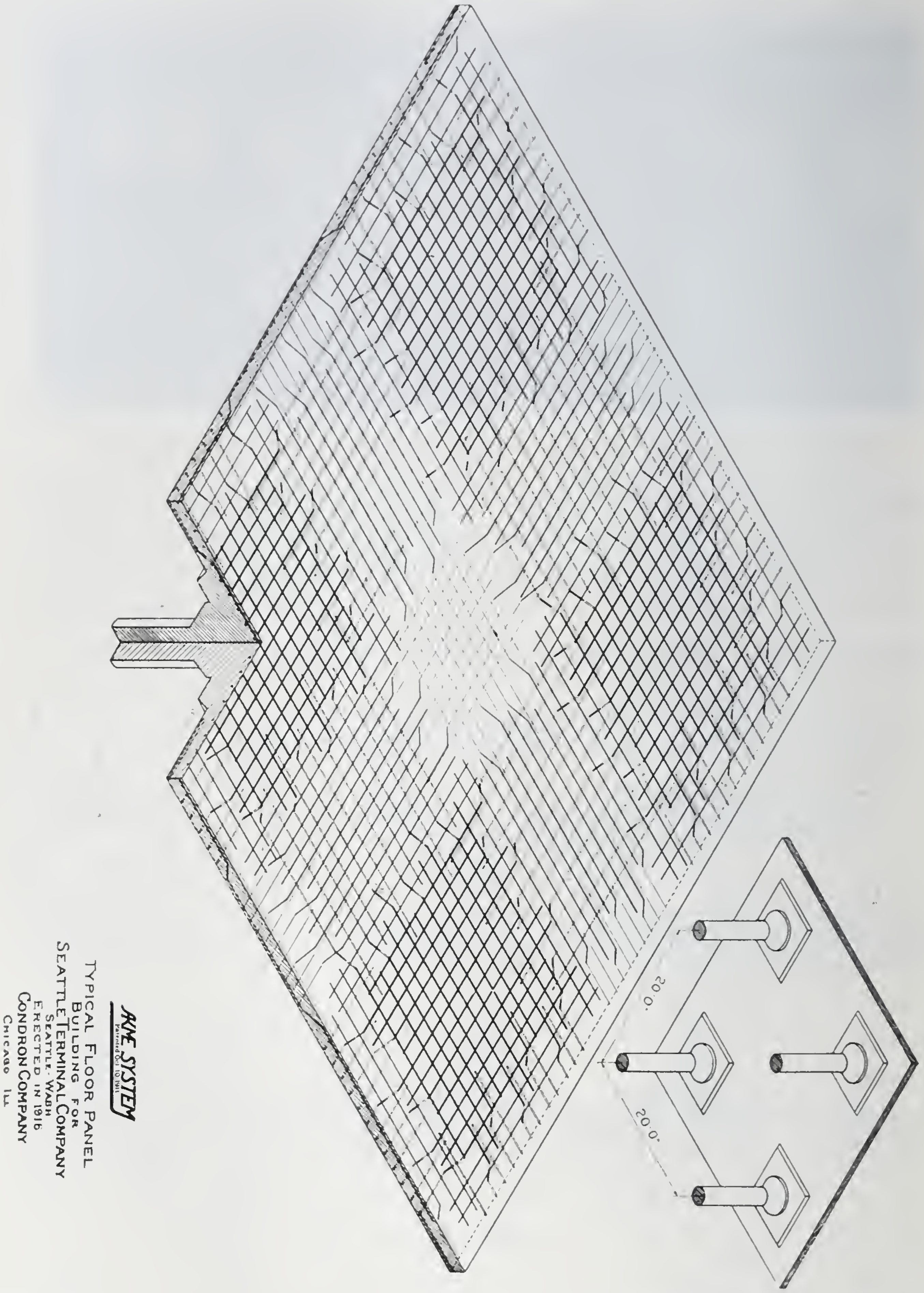


Fig. 5. Isometric Drawing Showing Reinforcement—Seattle Terminal Building.

inforcing bars was 18 700 pounds at the middle of the wall panel and 19 500 pounds over the wall column. Theoretically the unit

stress in the steel under this test should have been 36 000 pounds per square inch as the structure was designed for a unit stress in the steel of 18 000 pounds under designed load. The arrangement of the reinforcing of these flat-slab floors in the Sears, Roebuck Building at Seattle is shown in the isometric drawing, Fig. 5.

The test data which we have seem to indicate that the method we have used in designing exterior or wall panels is consistent, and in our opinion 20 per cent increase in the moments for interior panels is ample in determining moments for exterior panels. This should apply only to the belts perpendicular or diagonal to the wall and does not apply to the belts extending parallel with the wall.

The method of designing flat-slab floor construction which has been used in the writer's office is clearly illustrated by Fig. 6 which is a reproduction of a sheet taken from our regular office standards. This sheet shows the plan of typical square and oblong panels with the complete formulas for the same. It is our practice not to use a spandrel beam where a flaring head is permissible on the wall columns, unless there is an unusually heavy wall load. In the case of ordinary spandrels with windows we do not find it necessary to use a spandrel beam, provided we can use a flaring head on wall columns. The spandrel beam may be cast either above or below the upper surface of the floor. The writer believes that the omission of deep spandrel girders in connection with flat-slab floor construction originated in the office of his company, where this has been the practice from the first. It has frequently been necessary to provide spandrel beams in order to conform to architectural details even though there was no necessity for these beams so far as loads and stresses were concerned.

In Fig. 7 is shown a moment and steel reinforcement diagram for a typical flat-slab panel with a span of 22 feet. It will be noted that the location of ends and bends of bars in the slab is determined from the moment diagram in a manner similar to that of finding the lengths of cover plates for plate girders. From

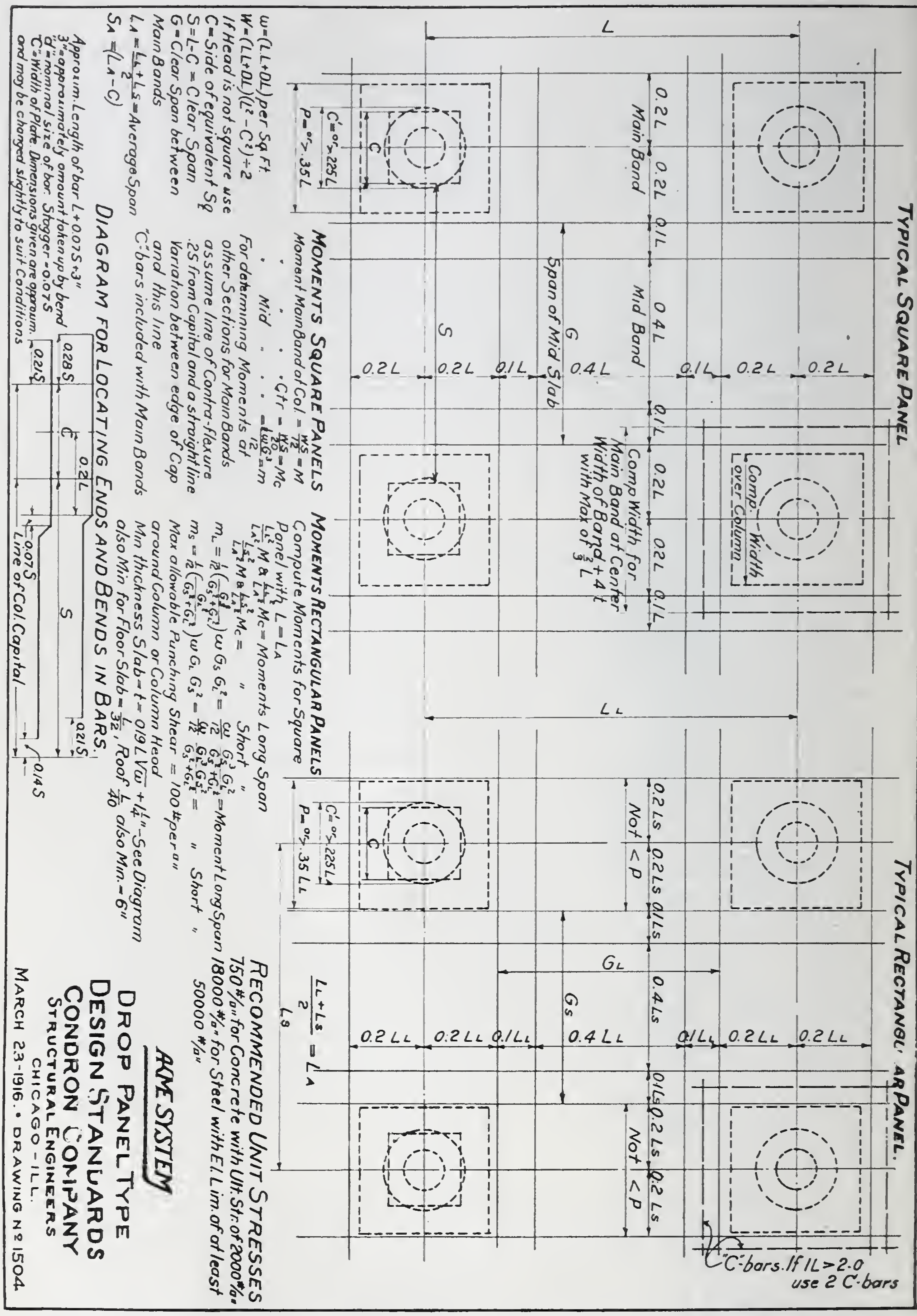


Fig. 6. Design Standard Sheet—Drop-Panel-Flat-Slab, Two-Way Type.

the diagram it is apparent that the method of placing bars makes ample provision for all tensile stresses in both the top and bottom portion of the slab.

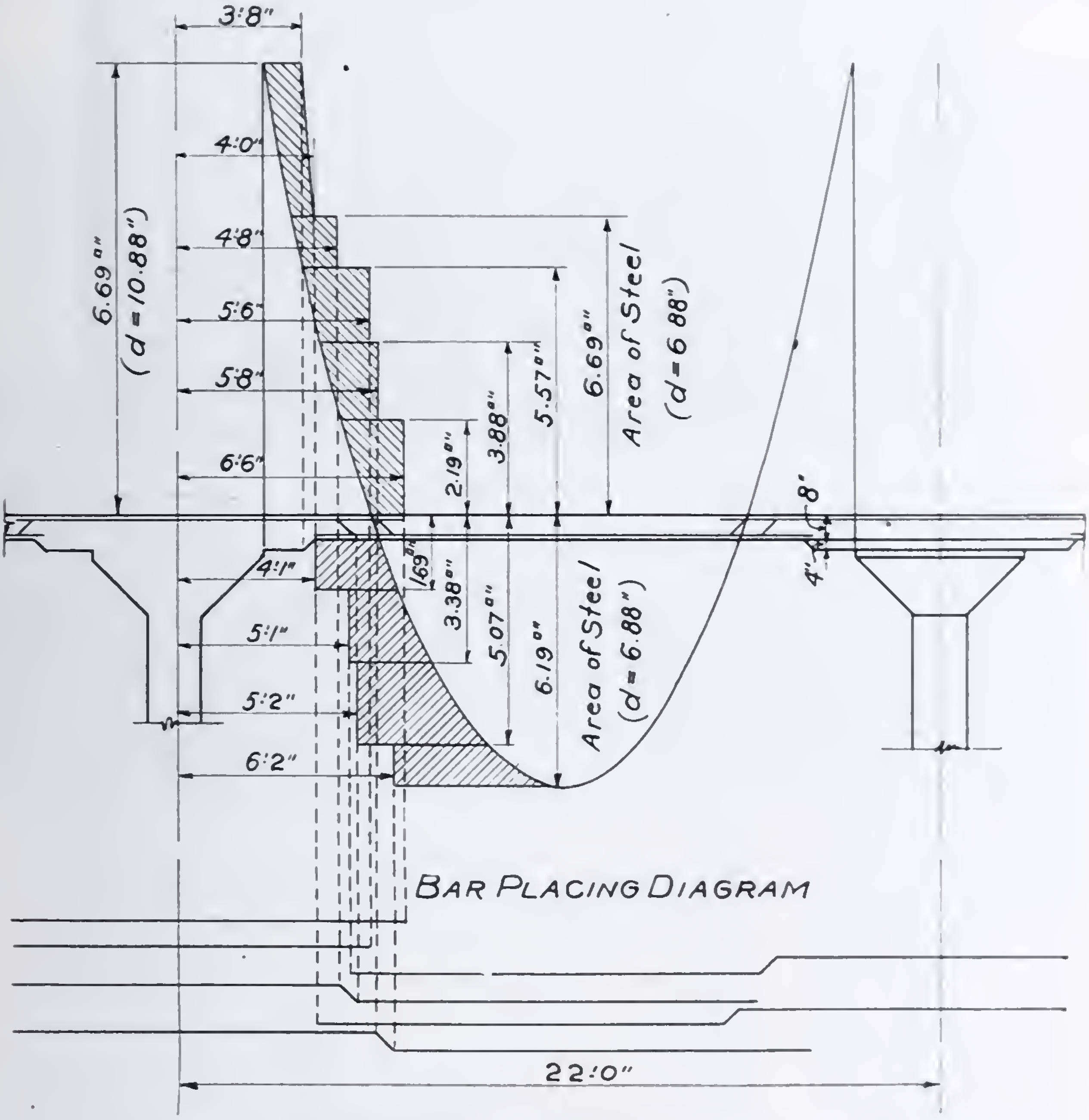


Fig. 7. Moment and Steel Reinforcement Diagram.

The "two-way" system of flat-slab construction, which was originated and patented by Mr. F. F. Sinks, who at the time was associated with the writer in the Condron & Sinks Company, was first used in an experimental model in 1907, and later in a building erected in 1908. Both of these earlier applications were to a flat slab without paneled ceiling, and the first application of this method of construction in which the ceiling was paneled—that is,

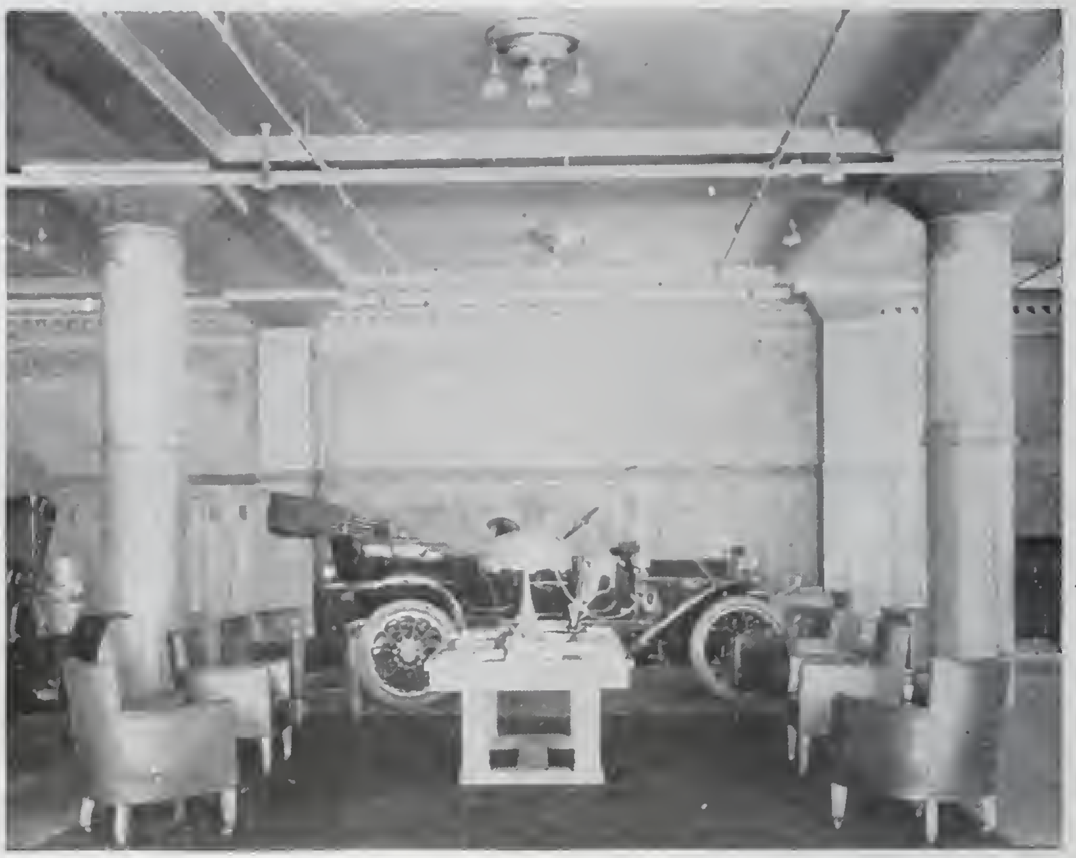


Fig. 8. Interior—Studebaker Building, Chicago—Paneled Ceiling Type of Construction.

Fig. 9. Interior—Wagner Electric Manufacturing Company's Building No. 8—Drop-Panel Type of Construction.

the slab in the central portion of the panel made thinner than the remainder of the slab—was in a building erected in 1909. A number of buildings have been erected with paneled ceilings but most of the construction built in accordance with the Sinks invention has been of the usual flat-slab type. The two types are illustrated in Fig. 8 and 9, while Fig. 10 very clearly shows this flat-slab construction without spandrel girders.

Fig. 10. Construction View—Wagner Electric Manufacturing Company's Building No. 8—Two-Way Flat Slab.

Mr. Edward Godfrey:* Some of the flat-slab theories are based on the theory of a flat plate of indefinite extent, supported on points, and loaded uniformly over the entire surface; and these theories include the assumption that the material of which the flat plate is composed is homogeneous and of equal strength and elasticity in all directions. Poisson's ratio is brought in as an accompainment, so as to reduce *ad libitum* the bending moment.

The flat slab in a building is not of indefinite extent; it is not supported on points; it is not, of necessity, uniformly loaded:

*Structural Engineer, Pittsburgh.

the material of which it is composed is not homogeneous and is not of equal strength or elasticity in all directions. In other words, there is not a single assumption of the flat-plate theory that is applicable to the flat slab. The homogeneity of the material and the application of Poisson's ratio, in particular, are quite foreign to the flat slab. There is therefore a vast amount of fine-spun and intricate theory, concerning flat slabs, that is worth quite a little less than nothing, for it clutters up the literature of engineering.

A word about Poisson's ratio. Simply stated, and applied to a concrete case, its theory is this: If a sheet of metal be pulled in one direction by a uniformly distributed load, the effect is to reduce the width of the sheet in the direction at right angles to this first set of forces, and presumably to cause compression, or the effect of compression, in that direction. Now, if a second set of forces were applied at right angles to the first set, the tension would, by this theory, be reduced by the amount of compression that is, presumably, caused by the first set, and, on the same theory, the effect of this second set is reciprocally reduced by the first set. It is a theory pure and simple. In the case where it would have its most perfect application, no engineer would think of designing in accordance with this theory—that is, in a hollow sphere under internal pressure.

Prof. H. T. Eddy is one of the chief exponents of the flat-plate theory as applied to flat slabs, and the value he assumes in Poisson's ratio would mean that by some means the reinforcing rods are pulled in a horizontal direction, normal to their length, with a force of about 8000 pounds per square inch. The only medium through which this force could be applied to the rods is the concrete, and the concrete could not do one per cent. of the job.

Another theory is one advanced by Mr. John Krippner, in *Engineering Record,* June 27, 1914, v. 69, p. 731. Mr. Krippner finds the bending moment in a ring around the column head. He then treats all rods crossing the column head, or anywhere near it, as though they were radial. Some of the rods, instead of being radial lie out near the border of the column head; some of them are clear outside of the circle; only a very small number

are anywhere near a radial position—and yet all of them are considered as though they were radial. Such absurd theorizing and such practice in designing ought not to call for serious comment. And yet it is just such theories that form the basis of flat-slab systems.

If a rectangular slab of span L be supported along two edges, the sum of the bending moment in the mid-section and that at either of the supporting edges is WL/8, W being the uniform load supported. A theory of flat slabs takes this demonstrable fact and seems to reason about like this: This bending moment WL/8 is the moment for one panel, hence if we provide for this moment somewhere in the panel, we shall satisfy the theory of flexure. The Pittsburgh ruling, about which more will be said later, is based on this sort of alleged reasoning. A section is taken around the column head and another, with the column as a center, midway between columns. All rods in the neighborhood are considered at right angles to these sections, and these sections, with their alleged reinforcement—some of which does not even touch them—are supposed to resist "WL/8" for that panel. It is as though all one had to do were to find rods somewhere, which, if assumed to be somewhere else, would resist bending moments that exist still somewhere else.

Mr. John R. Nichols, in the *Transactions* of the American Society of Civil Engineers, 1914, v. 77, p. 1670, showed what the writer had pointed out in the *Engineering News*, February 29, 1912, v. 67, p. 403—namely, that there are perfectly definite static limitations to the bending moments in the main sections of a flat slab. The values of these static limitations are easily deduced by the simplest theory, and their correctness cannot possibly be questioned. This is the only correct and safe theoretical treatment of the flat slab, though it is not in any favor with commercial designers (Fig. 11). The writer has repeatedly set forth the proposition that in Fig. 11 the numerical sum of the bending moment at C D and that at A B (or E F) is W′L′/8. If, then, both sections are reinforced alike, the absolute minimum bending moment is W′L′/16. And the flat-slab design that does not meet this criterion in the matter of reinforcing steel used, must of

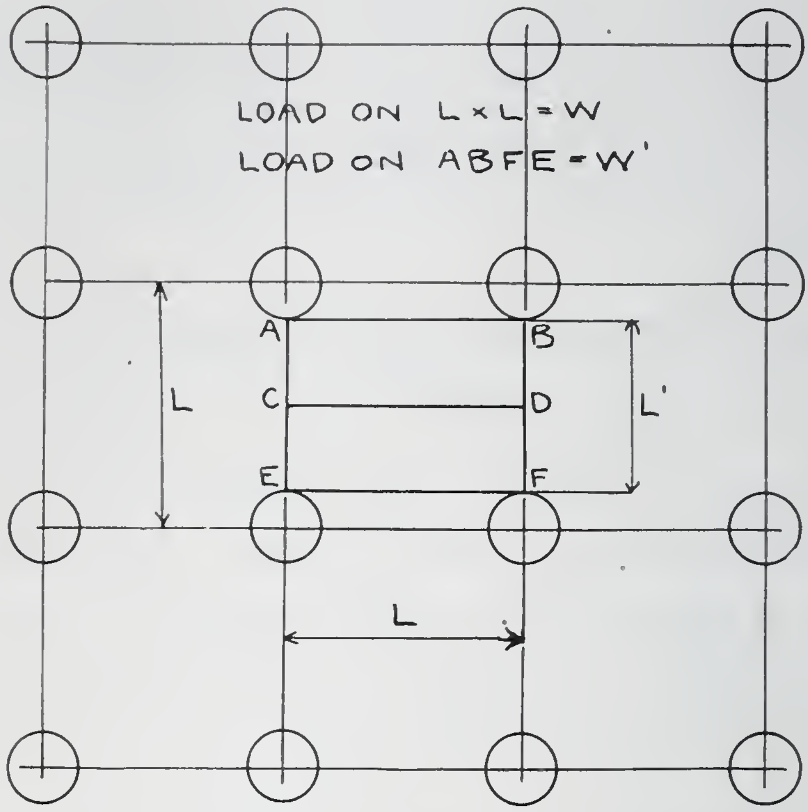


Fig. 11. Showing where Static Limitation Has Application to Flat Salb.

necessity be dependent on tension in the concrete by just the amount that it falls short. And some flat-slab designs are very far from meeting this criterion.

In the *Proceedings* (*Papers and discussions*), of the American Society of Civil Engineers, v. 43, p. 887, Prof. H. T. Eddy in criticizing the Joint Committee report says:

"Any assumption of the validity and applicability of statical analysis to continuous flat slabs is incorrect and leads to erroneous results just as much as in the case of continuous beams or of any other indeterminate structures.

"Any structure in which the magnitude or distribution of the stresses in any part of it undergoes any alteration by varying the rigidity of any of its members or elements is statically indeterminate. Hence the principles of statics cannot be assumed to be applicable to such a structure unless there is definite proof that the statical principles sought to be applied are actually valid for the case in hand, so that it is beside the mark to adduce any statical limitations or requirements in flat slabs, such as are adduced in the report, because statical principles must here be subordinated to the principle of rigidities, which is the guiding principle in all indeterminate structures."

Then the astounding part of Professor Eddy's argument is that he goes on to prove (p. 892) that there is in reality a static

limitation to the bending moments in a flat slab, but he adroitly splits it up so that it appears to be one-half of the real value.

Professor Eddy further states:

"It is a well-known proposition of beam theory that the sum total of half the numerical values of the applied negative bending moments at the ends of any span of length L plus the numerical value of the positive bending moment applied at mid-span, all arising from a uniformly distributed load, W, amounts to WL/8, whether the span is a simple one with end supports merely or is continuous, or partly so, and this without regard to the loading or absence of loading on other spans of the beam. This proposition holds true regardless of the relative magnitudes of the moments of inertia of the beam at its successive cross-sections by which the relative rigidities of the beam in resisting applied bending moments is expressed. It likewise is independent of the rigidity of the supports or their connections with the beam in resisting applied bending moments."

The foregoing quotation states mathematical verities, but they are exactly contrary to what Professor Eddy says regarding static limitations. In the paragraph just quoted he shows that there is an absolutely definite limit to the bending moments in a continuous beam. The same is true of any continuous slab, and Professor Eddy goes on to apply it to the flat slab (Fig. 12). Professor Eddy says in effect that because the shear on J K L M is W, therefore that on K L is W/4, and therefore slab E F G H can be treated as a slab loaded with W/2. This is so utterly absurd that it ought not to be given a place in engineering literature. But there it stands in the *Proceedings* of the American

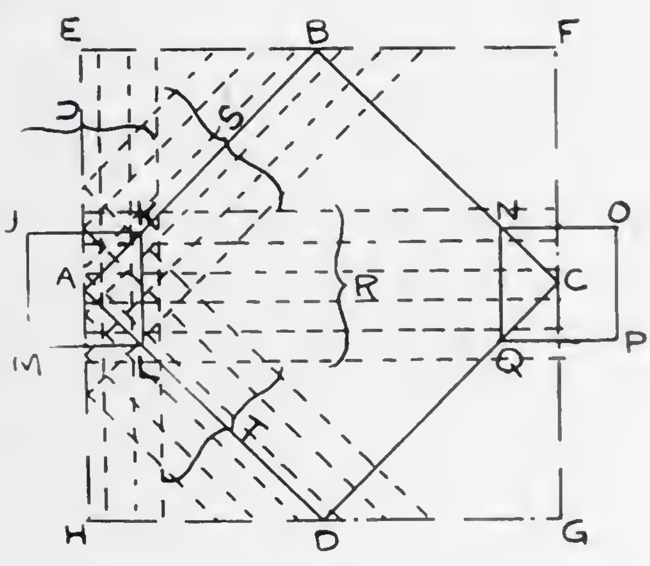


Fig. 12. Showing where Professor Eddy neglects Large Bending Moments in His Theory.

Society of Civil Engineers, while the writer's attempt, as a member of the Joint Committee, to show up Professor Eddy's error was rejected by the Publication Committee.

To answer Professor Eddy briefly: section K L is a section in slab A B C D and though this may be considered as carrying a load W/2, there is a negative bending moment in section A C which he ignores entirely. The only reinforcement against this negative moment consists in half the rods S and half the rods T with half the rods U. But all of the rods S, R, and T are already accredited to section K L by Professor Eddy.

On one point the common theories of flat slabs are silent. This concerns exterior columns. A flat-slab system is essentially a cantilever system. It derives much of the strength exhibited in tests from the balancing of loads over column heads; but when it comes to exterior columns, where the cantilever stresses are greatest, some designers actually make these columns weaker, concerns exterior columns. A flat-slab system is essentially a relatively, than interior columns, and few theories take any account of the bending moment in these columns. In this connection appears one of the most astounding statements that the writer has seen in recent years. It is quoted from Professor Eddy's above-mentioned paper, page 891.

> "These identical deformations will ensure nearly equal shears and bending moments in outside and inside panels, because the saucer-shaped deformations will be nearly the same in each."

Thus the question of bending moments in exterior panels and exterior columns is nonchalantly waved aside. In a professedly cantilever system, at the very place where the cantilever action has the most destructive effect, the question of the bending moments due to this action is utterly ignored, because, forsooth, "the saucer-shaped deformations will be nearly the same in each."

Tests are cited as confirming the spurious theories of flat slabs. And there have been some carefully guarded tests that have shown high results. There have been scarcely any tests made that are really critical for the flat slab, and there has been a large amount of false interpretation of tests.

If, as Professor Eddy claims, an exterior panel is as well conditioned as an interior panel, an isolated panel ought to be as

good as either. And yet no fair test has ever been made on an isolated panel. A test reported by Professor Eddy in the *Engineering News,* March 27, 1913, v. 69, p. 624, on an isolated panel was made with short chunky columns of a diameter one-eighth of the side of the panel. The panel had an overhang, which was carefully loaded to balance the interior load, and the loading material was partly long rails, which could in no wise simulate a uniform load. Furthermore, tests have not been made on a row of exterior panels supported by columns. The nearest thing to this is the Bell Street Warehouse in Seattle, where both the slab and the columns failed.

Exterior bays have been tested where the outside support was a deep girder. Interior bays have been tested where the surrounding bays were idle. It is well known that a continuous slab of any kind whatsoever is greatly aided in supporting its load by the dishing effect when the surrounding construction is idle.

The writer tested the middle panel in a building three panels wide, in a structure of the beam and slab type. The results were excellent. He then proceeded to load the other two panels across the building. The result was practical failure.

Some tests have been made in which the steel stresses have been measured, and then, because these stresses come near agreement with the supposed flat-slab theories, the theories are asserted to be correct. Now none of these theories take any account of tension in the concrete and the substantial aid that it contributes toward the support of the load. It can be readily shown that in a whole slab the tension in the concrete will be five or six times as much as that in the steel. *Hence a theory that ignores the tension in the concrete is completely discredited, if in a whole slab the tension, of the theory, is found to exist in the steel reinforcement.* On the other hand the true theory of a flat slab will indicate that there are possible stresses in the steel that will not be realized in the ordinary test. This gives rise to the charge that such theory is not indeed correct because it does not work out in tests. But take a structure and make a real test on it. Load a complete row of bays from one side to the other so that the slab will take a cylindical shape and not a saucer shape. Let the load be applied on the roof or the top story where the columns

are not enormous in size. And let the load be double the design load. Until such test is made, and until it disproves the theory, no one has the right to say that the theory of statical limitations is discredited. Any owner has the right to load his building in this way.

It is tension in the concrete that supports the greater part of the load on any flat slab, and any theory that ignores this fact is a false theory. Furthermore, any system of design that takes advantage of this tension and reduces the amount of steel to only a fraction of that which would be required to take that tension, should the concrete crack, is a system that ought not to be countenanced by engineers.

Pittsburgh has a ruling for flat-slab design which was published in the *Engineering News,* May 11, 1916, v. 75, p. 910. So far as the writer can discover this ruling has no legal sanction. It is directly contrary to the existing law, which makes it mandatory on the Bureau of Building Inspection to base any such ruling on tests to destruction that show a factor of safety of four. No tests have ever shown any such results. In fact the Bell Street Warehouse in Seattle, a building agreeing with this ruling in interior panels, showed practical failure at a trifle above the supposed safe load for exterior panels, and dangerously high stresses for interior panels.

The Bell Street Warehouse was a flat-slab building of the Turner standard design. It was tested by the owners and not by the builders. A little more than the alleged safe load was placed on the floor. Outside panels were loaded (the outside support was columns and not a wall or girder). One of the engineers engaged on the test informed the writer that they were afraid to carry out the full program of tests on account of the dangerous results anticipated from a partial test. The steel and concrete stresses measured in the building were in many cases enormous and dangerous. The patentee excused the failure on the plea that the concrete had not had time to season. The concrete was 113 days old when the tests began. (See *Engineering Record,* May 13, 1916, v. 73, p. 647.) A re-test was later made on this

building, and it confirmed the results of the first test. The patentee's explanation then was that rods which were measured for stress must have been kinked between the measuring points.

It is to be observed that the Seattle tests and failure confirmed the true theory of the flat slab.

A few years ago the writer checked the design of a building being constructed in Pittsburgh. The building was presumably designed on the Pittsburgh ruling and its design had been passed by the Bureau of Building Inspection. It was designed precisely on the same standard as the Bell Street Warehouse, with exterior bays having the same reinforcement as interior bays, and with exterior columns of the rodded variety. Instead of bending being provided for in these exterior columns to take the cantilever stresses, the exterior columns *were not good for one-half of the direct load on the plain requirements of the Pittsburgh Building Code.* The columns were quickly reinforced with hooping when the latter fact was demonstrated, but the writer's protests in other respects were ignored.

The Pittsburgh ruling now requires additional reinforcement in exterior panels, thus departing from the Turner standard. For interior panels it is the Turner standard, and that standard is the lowest in the scale of commercial standards of which the writer has any knowledge. A large building in the Southwest after many years of use, sagged several inches in the floor panels and had to be vacated. It was a Turner standard design.

The Joint Committee on Concrete and Reinforced Concrete, of which the writer is a member, recently issued a report in which a standard for the design of flat slabs is given. The reinforcement required in this standard approaches that which would be required by the true theory of statical limitations, as set forth by Mr. Nichols and the writer, and therefore depends but little on tension in the concrete.

It is significant to note that the Joint Committee report requires 53 per cent. more reinforcement in mid-panel sections and 135 per cent. more reinforcement in the column-head section than does the Pittsburgh ruling. It is significant also to compare the Pittsburgh ruling for flat slabs with the Pittsburgh Building Code for beam-and-slab type floors. Fig. 13 will serve to show this

comparison. The Pittsburgh ruling is based on the hypothesis that the resistance of section M N O P added to that of A B C D (or circle I J K L, either satisfies the wording of the ruling)

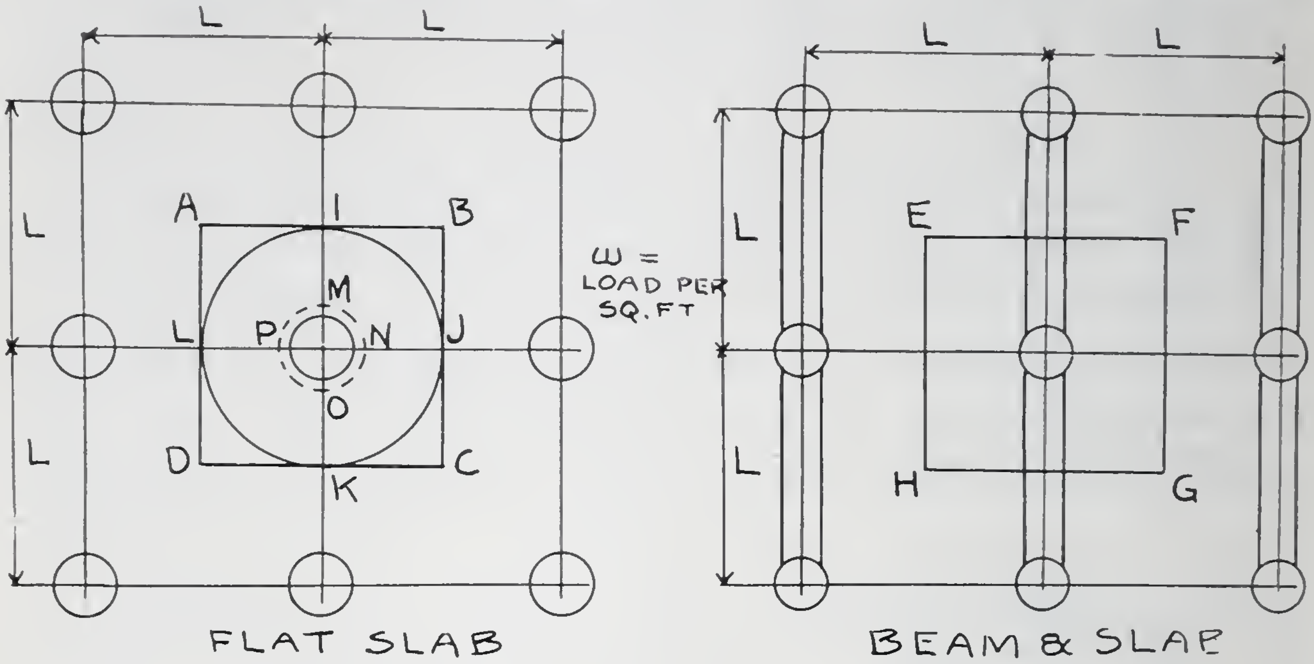


Fig. 13. Comparison between Flat-Slab and Beam-Slab Standards.

should be wL3/8. There is no principle of statics that either demands this or is satisfied by it. Besides this gross error, two sections are assumed around the column head—one cylindrical for figuring the resisting moment, and the other conical to reach out and tag some rods clear outside of this cylinder. Then all rods within and without the cylinder are considered radial, another gross error. Now in the square A B C D the Pittsburgh ruling requires a resisting moment 0.0625 wL^3, or since half the rods are cut at 45 degrees and all are counted as normal, the effective resisting moment is 85 per cent. of this, or 0.0531 wL^3. In the corresponding section E F B H of the beam-and-slab construction the Building Code requires that the girder, which is cut by E F and G H, be designed to resist 0.083 wL^3 in *each* section, or 0.1667 wL^3. Also the slab, which is cut by E H and F G must be designed to resist the same moment in each section, or 0.1667 wL^3—a total of 0.3333 wL^3—more than six times the strength demanded of the flat slab by the Pittsburgh ruling and on the basis of its derivation. Here is something for engineers, particularly Pittsburgh engineers, to ponder. It demonstrates at once the inadequacy and injustice of the Pittsburgh ruling, and the totally erroneous hypothesis upon which it is based.

There is no doubt that if the beam-and-slab construction had behind it business organizations and a licensing system, tests—spot tests and strip tests—could be accumulated, which would convince many that requirements of building codes could be shaded so as to give that construction a better advantage in the building market.

NECROLOGY.

Albree, Chester B.
Bigelow, Edward Manning
Bradford, William
Bryan, James
Cordes, John W.
Coulter, Alexander
Dean, Ellsworth
Engstrom, Frans
Fitzgerald, Charles
Foller, Charles S.
Frost, Albert E.
Fulton, Louis B.
Gilfillan, George A.
Haines, John Locke
Koch, Walter E.
Lovejoy, Francis F.
Macbeth, George A.
MacRoberts, James H.
Monroe, William T.
Orr, Robert Sherrard
Ow, Henry S.
Ramsey, Joseph H., Jr.
Schluederberg, George W.
Swan, Robert
Yardley, Edmund

Chester B. Albree

Pittsburgh, April 8, 1862.
Pittsburgh, May 27, 1916.
Director, 1900-1901.
Vice President, 1902.
President, 1903.
Joined the Society, March 1888.

Born in Pittsburgh on April 8, 1862, of an old Pittsburgh family, Mr. Albree spent his boyhood in Pittsburgh, but later went to the Worcester Polytechnic Institute in Worcester, Mass., where he graduated in 1884. He almost immediately entered into the ornamental iron business and established the Chester B. Albree Iron Works, in which he continued until the time of his death.

While his principal business was ornamental iron, he started to manufacture pneumatic compression riveters about 1900 and did all the designing of these machines himself. He originated the very successful universal bail, whereby a suspended machine

may be turned in any position by merely swinging it through a bail so constructed as to keep the center of gravity of the machine always at the same height and thus preserve stable equilibrium. He took out several other valuable patents among others one covering an automatic pneumatic compression riveter, entirely doing away with the adjustment screw.

Mr. Albree was probably most widely known as a manufacturer and designer of bridge railing and many beautiful designs of his can be seen in all parts of the United States. Perhaps one of the best is that along the upper end of the Riverside Drive in New York City.

Mr. Albree always took a great interest in engineering and was a past president of the Engineers' Society of Western Pennsylvania. He was also a member of the American Society of Mechanical Engineers and of the American Association for the Advancement of Science and took an interest in the Pittsburgh Economic Club. He was a member of many business and social clubs among others the Duquesne Club and the Pittsburgh Club.

Mr. Albree married Mary Phillips Lyon of Hartford, Conn., but had no children. He died on May 27, 1916, after an illness of several months. He is greatly mourned by his friends, for he was a versatile, many sided man and a most delightful companion.

Edward Manning Bigelow

Pittsburgh, November 6, 1850.
Pittsburgh, December 6, 1916.
Joined the Society, January 1880.

Edward Manning Bigelow was born in Pittsburgh, November 6, 1850. He received his early education in the public schools, and after graduation entered the Western University of Pennsylvania (University of Pittsburgh) to take up the study of Civil Engineering.

Upon leaving college, he entered the service of the City of Pittsburgh on the staff of the City Engineer, which position he held until he was promoted to engineer in charge of certain street

construction. He left the city for a short time and was re-employed as Assistant Engineer in charge of surveys and location of streets from 1876 to 1878, and in charge of construction from 1878 to 1882.

On January 9, 1882, he was unanimously elected City Engineer by Council, which position he held until 1885, and from 1885 to 1888 he was City Engineer and Commissioner of Highways. About this time the City Charter was changed, the Department of Public Works being created, and Mr. Bigelow was appointed Director of the Department of Public Works, which office he held for two full terms of three years each.

On June 1, 1911, the Governor of Pennsylvania, John K. Tener, appointed Mr. Bigelow to the office of State Commissioner of Highways, which office he held until April, 1915. He was honored by a re-appointment to the office of Director of the Department of Public Works of Pittsburgh just prior to his death.

Mr. Bigelow was an engineer of broad vision and foresight; his plans for improvements were always comprehensive, and provided well for the future. Under his supervision, many miles of Pittsburgh's streets and sewers were laid out and constructed, including the boulevards. His greatest achievement was the development of Highland, and Schenley Parks. It was also under his administration that work on the water purification plant was started. He was known as the "Father of the Parks."

Mr. Bigelow during his term of office as Commissioner of Highways directed the making of plans for the improvement of the state highways in anticipation of securing funds for their improvement and much credit is due him for the work performed with the funds at his disposal.

Mr. Bigelow took an active interest in financial, commercial and social activities. He was President of the Liberty National Bank, the Liberty Savings Bank and the Homewood Cemetery. He was Trustee of Carnegie Institute of Technology from the time of its founding and took an active interest in the development of this institution. He was also a trustee of Andrew Carnegie's other benefactions—the Carnegie Library, the Carnegie Institute and the Carnegie Hero Fund Commission—and of the Schenley Memorial Commission. He was a member of the East Liberty

Presbyterian Church; also of the American Society of Civil Engineers and of the Engineers' Society of Western Pennsylvania.

James Bryan

Preston, England, October 13, 1861.
Pittsburgh, February 20, 1918.
Joined the Society, November 1914.

Mr. Bryan received an education in mechanics and engineering in his youth in England. His adoption of engineering was natural, he being a descendent of a long line of mechanics and engineers. He came to the United States in the year 1888, becoming associated with the Corliss Engine Company of Providence, R. I., where he remained four years. His next association was with the Thompson-Houston Company of Lynn, Mass., from where he was transferred to the Edison General Electric Company of Schenectady, N. Y.

Mr. Bryan came to Pittsburgh in the year 1896 as Engineer for a number of the traction lines forming part of the present system, in the development of which he played a prominent part.

He began private practice as Consulting Engineer in 1898. He became widely known for his many achievements in engineering, among which probably the most notable was his conception and practical achievement of higher direct-current voltage for traction purposes, especially for interurban conditions. His first installation of the idea was the construction of the Pittsburgh, Harmony, Butler & New Castle Railway, the result of which led to the rapid use of higher voltage direct current for both interurban railways and trunk line electrification.

Mr. Bryan was a member of the Chamber of Commerce, the Union Club, the Country Club, and the Engineers' Society of Western Pennsylvania. His widow, Mrs. Agnes Bryan; five daughters: Mrs. S. L. Roush, Mrs. A. R. Cancelliere, Mrs. C. K. Sheridan of Cleveland, Ohio, Mary and Florence at home; and two sons: Joseph and James, Jr., survive.

Charles Fitzgerald

Monroe, Orange Co., N. Y., October 1, 1859.
Pittsburgh, June 2, 1917.
Joined the Society, January 1896.

Charles Fitzgerald was born in Monroe, Orange County, N. Y., October 1, 1859, and received his common school education in the school of that village. His early mechanical training was with the Ramapo Car Wheel Company, with which company he was connected from 1879 to 1882, leaving their employ to take a position with John Roach & Sons, shipbuilders, Chester, Pa. His next position was with the American Shipbuilding Company, Philadelphia, Pa., where he was foreman in charge of the erection of marine engines. From 1885 to 1889 he was connected with Robert Wetherill & Company, engine builders of Chester, Pa., as outside erection engineer.

It was while acting in this capacity that he came to Pittsburgh where he was destined to firmly establish himself and make his final home. He superintended the construction of the power-plant for the operation of the cable line of the Citizens Traction Company, and, when the plant was ready for operation, he was persuaded to take charge of it and the cable lines as Chief Engineer. When this system was converted to use electricity as its motive power, it became a part of the Consolidated Traction Company, Mr. Fitzgerald becoming chief engineer of all power-plants. In a very short time he was promoted to the position of General Superintendent, which position he held until the system with which he was connected, together with all other street railway lines in Pittsburgh, was leased to the Pittsburgh Railways Company, at which time he accepted the position of mechanical engineer with Booth & Flinn, Pittsburgh. In 1906 became General Manager of the Pittsburgh-Brazilian Dredging Co., spending about three years in the interior of Brazil.

In 1909 he accepted a position with the Pittsburgh, Valve, Foundry & Construction Company in a sales and engineering capacity, being Superintendent of Construction for the Baltimore High Pressure Fire System, installed in 1910-1911. At the time of his death he was Assistant to the President of this company.

He became a member of the Engineers' Society of Western Pennsylvania in 1896 and of the American Society of Mechanical Engineers in 1912. He was also a member of the Engineers' Society of Pennsylvania.

He died of pneumonia at his home in Pittsburgh on June 2, 1917.

Mr. Fitzgerald was of a quiet and unassuming nature and that he was grateful to fill the various posts of duty to which he was called, is evidenced by the fact that he constantly progressed as he went through life. He was a loyal friend, and as a member of the Society, he was always devoted to its welfare.

Dr. Albert Ellis Frost, A.M. Sc.D.

St. Johnsbury, Vt., August 9, 1851.
Pittsburgh, May 11, 1917.
Treasurer, 1881-1917.
Charter Member, 1880.

Dr. Albert Ellis Frost was born in St. Johnsbury, Vt., on August 9, 1851. His father was Selim Frost, and his mother Emily Ellis Frost, both of English stock. He received his early education in the public schools, and prepared for college in St. Johnsbury Academy. He graduated from Dartmouth College with the degree of A.B. in 1872, receiving his Master's degree in 1887, and the degree of ScD. in 1897. After graduation at Dartmouth, he became assistant to Prof. S. P. Langley, Director of the Allegheny Observatory, and acquired a great insight into the modern theory of optics, and also served with that famous scientist and inventor in developing the first heavier-than-air flying machine, until 1875, when he became Professor of Physics in the Pittsburgh Central High School, which position he filled with distinction for ten years. In 1885 he accepted the chair of Physics in the Western University of Pennsylvania, now the University of Pittsburgh, and in 1888, in addition to his other duties, he became Registrar, which position he held at the time of his death. He also served as Treasurer of the University from 1892 to 1909,

and as Acting Dean of the School of Engineering and the College, and Professor of Mathematics from 1908 to 1910. Dr. Frost was a charter member of the Engineers' Society of Western Pennsylvania, and also its only Treasurer until the time of his death. He was President of the Academy of Science and Art from 1902 to 1905, and also a member of the Executive Committee of the University Extension Society.

Dr. Frost was known to his students and colleagues primarily as a man of most charming personality. This single gift alone does not endear a teacher to his pupils, but it is necessary to combine, as Dr. Frost did, this characteristic with real ability to win both the regard and affection of his students and colleagues. He was always modest in regard to his accomplishments in his chosen fields of science as in everything else, and in recent years devoted much of his time and energy to the personal welfare of the students and the University. The unassuming greatness of Dr. Frost will stand recorded in the annals of scientific accomplishment, and his nobility ever remain sacred in the memories of his associates.

Louis B. Fulton

Beaver, Pa., October 19, 1841.
Pittsburgh, April 7, 1916.
Joined the Society, April 1888.

Louis B. Fulton, late President of The Chaplin-Fulton Manufacturing Company, died at his home in Pittsburgh, on April 7, 1916, in the seventy-fourth year of his age.

Mr. Fulton had been President of the company since its organization as a corporation in 1891, and previous to that had been President of The Chaplin-Fulton Company, Ltd., during the period of its existence. He was an accomplished mechanic and engineer, thoroughly versed in the laws of pressure and its control, and a pioneer in the development of means for this purpose. He was a fertile inventor and his genius and skill enabled him to take out many important patents.

About 1882, when the natural gas business was just entering upon its wonderful career and when its votaries were groping in the dark for some appliance suitable to handle and control the enormous gas pressures developed, Mr. Fulton invented the famous Fulton was regulator. His ingenuity and pains taking application enabled him to perfect it so that it soon became the recognized superior of all appliances for this purpose and is known and used all over the world wherever natural gas is used. It was stated by a distinguished engineer at a meeting of the Natural Gas Association of America, that Mr. Fulton was one of three men who by their inventions had made possible the practical use of natural gas.

Mr. Fulton was also an expert investigator of the laws of distribution of water and steam and had demonstrated his knowledge by many inventions along these lines. He was often consulted by various managers of plants, who valued his judgment, and he gave gratuitously the benefit of his knowledge and experience.

Mr. Fulton was a delightful friend and companion, with a large circle of friends, who will mourn his loss. In manner, he was modest and unassuming. He was a self-made man. Beginning life as a poor boy, by his own unaided efforts he acquired a practical education and rose to a position of eminence as a mechanical engineer and manufacturer. A complete story of his long and honorable career would prove a living inspiration to the youth of our country.

JOHN LOCKE HAINES

Saco, Maine, May 16, 1866.
Pittsburgh, December 14, 1917.
Joined the Society, January 1904.

John Locke Haines, for some years Assistant to Willis L. King, Vice President of the Jones & Laughlin Steel Company, and himself a director of the company, died on Friday, December 14, 1917, in Pittsburgh, after an illness of nearly a year. Mrs. Haines survives him.

Mr. Haines was born in Saco, Maine, May 16, 1866, and received his early education in the public schools of Portland, Maine. At the age of 21, he entered the employ of the Pepperell Manufacturing Company at Biddeford, Maine, manufacturer of cotton goods. Seven years later Mr. Haines went to Boston and for three years was connected with Wallace R. Morse & Company, commission merchants. He came to Pittsburgh in 1897 and entered the service of the Jones & Laughlin Steel Company. He was first employed in the Structural Sales Department and two years later was placed in charge of the Order Department. He was appointed to the position of Assistant to the Vice President on January 1, 1910.

Mr. Haines had been a member of the Engineers' Society of Western Pennsylvania since January, 1904.

He was an earnest member of this Society and took a very active interest in all that pertained to engineering. His genial manner endeared him to all his friends and associates.

Walter E. Koch

Breschow, London, England, March 31, 1848.
El Paso, Texas, May 25, 1916.
Joined the Society, April 1887.

Walter E. Koch was born March 31, 1848, at Breschow, part of London, England. He was educated at Marlboro College, Wellshire, and St. Johns College, Cambridge, where he took his M.T. in Natural Science. He worked for Sir William Siemens at Landore, Wales, from 1876 to 1879, perfecting the Siemens (open-hearth reversing) furnaces, which are now used everywhere. He went from Swansea to the Parkhead Iron Works, Glasgow, Scotland, 1880-1881. In 1882 he was made manager of a mine near Stavanger, Norway. In 1883 he returned to Sir William Siemens to make some alterations in the Siemens furnace.

In 1884-1885 he left England, traveling through the United States to the mining centers of New Zealand and Australia for the Pyrites Smelting Company, of London, England.

In 1886 he came to Pittsburgh and was manager of the Spang Steel and Iron Company, Etna, Pa., until 1897, when he was made General Manager of the Lustre Mining Company, a Pittsburgh Company with mines and reduction works at Magistral, Durango, Mexico. He retained this position until 1904 and his work there along the line of endeavoring to develop a method of pyritic smelting for treatment of the ores, attracted much attention at that time. In 1905 he moved to El Paso, Texas, and was engaged in general consulting practice from that time until his death. In 1912 he became interested in the development of tin

mines at Terlingua, Texas, and was engaged by the Chisos Mining Company of Chicago to take the management of their operations there.

He was a member of the Iron and Steel Institute, England; the American Iron and Steel Institute; the American Institute of Mining Engineers; the Engineers' Society of Western Pennsylvania and other professional organizations. He will be remembered by the older members of the Engineers' Society of Western Pennsylvania for his frequent contributions to the discussions, which were always interesting and valuable.

He was married in 1889 to Mary Coyle, daughter of R. M. Coyle of Sharpsburg, who with his daughter Ellen and two brothers Arthur H. and E. C. Koch, survive.

He was a noble Christian and his prayer was "Make me a useful man, a Christian gentleman and an honor to Thee—Give me tact and a right judgment."

George Alexander Macbeth

Urbana, Ohio, October 29, 1845.
Pittsburgh, February 11, 1916.
Charter Member, January 1880.

In the death of Mr. George Alexander Macbeth, which occurred in this city February 11, the Engineers' Society of Western Pennsylvania has suffered the loss of one of its most highly honored members. He was numbered in the little group of able men who met in 1880 to found the Society and his name has ever since been borne upon the rolls.

Mr. Macbeth was born in Urbana, Ohio, October 29, 1845, the son of James Reed Macbeth and Francis Ann Bayard Macbeth. He had a good schooling experience at various institutions prior to his arrival in Pittsburgh in 1862 to engage in business. What difficulties he overcame and how, after 1862, he labored to perfect himself in his chosen field of glass making, we are not told, but we may be sure that it required an inflexible

purpose and diligent effort on his part to emerge, as he did in 1872, as a glass manufacturer.

Almost from the start he commenced those improvements in the composition and manipulation of glassware used for lighting purposes, which at once attracted attention and brought to his name a fame which spread all over the country and eventually reached all regions of the globe where American refined oils were used.

When one thinks of this illimitable field of enterprise—true not monopolized by any one firm, but having regard to the commercial value of a name—there was here offered to Mr. Macbeth an opportunity for becoming the head of a legitimate trust of vast capitalization. With him, however, the advancement of personal wealth was secondary to his desire to explore ways for advancing knowledge in his chosen sphere. The inauguration of the Carnegie Institute in this city with its department of Science, Art and Literature brought together a group of the best equipped minds and the most noted scientific workers of western Pennsylvania. Mr. Macbeth was intimately associated with the men of this group and enjoyed their esteem, as he did also that of Mr. Carnegie, to the fullest extent. Besides Mr. Macbeth's interest in applied science, he was an authority and critic of no mean rank on etchings and engravings. He was very active in bringing about the vast extension of the great building and especially the enlargement of the library and the establishment of its branches throughout the city.

When Mr. Macbeth's attention was directed to the government coast lighthouse service, he suggested possible improvements on the French and German lenses, not only in the composition of the materials but also in precision of workmanship, conducive to a higher efficiency and greater visual range of the light. He was at once invited to try his hand in this field.

To advance upon the methods of the highly trained and skilled workers of Europe was no trifling matter, as we may judge from what he himself said in a brief paragraph in his paper on "Light House Lenses," appearing in our Proceedings of April 1914, p. 243:

"In commencing the manufacture of these glasses, it seemed like assuming a duty unknown to investment as well as unknown loss or profit, but it also answered a challenge to an old glass center like Pittsburgh to produce high grade articles and run the risk in the endeavor. The sequel proves that in the membership of our Society is the spirit and perseverance to produce anything wanted in this field of optical glass."

In the January 1908 PROCEEDINGS of the Society, Mr. Macbeth contributed a brief "History of Glass Making." While many so-called "histories" of this fascinating art are in print, readers of Mr. Macbeth's all too brief story cannot fail to observe the mind of a clear-headed, practical and scientific glass worker, one amply qualified to interpret ancient records and separate the grains of truth from the "chaff" in which some writers envelop their ideas.

Besides glass for lighthouse lenses, Mr. Macbeth furnished some fine glass to Dr. Brashear, which the latter employed in making the range finders for our coast fortifications during the Spanish-American War. The Macbeth-Evans Glass Company also furnished Dr. Brashear glass for various medium-sized telescopic lenses, which were cut, polished and mounted at the well-known astronomical instrument establishment on Observatory Hill.

Mr. Macbeth chose the town of Charleroi, on the Monongahela River, 42 miles above Pittsburgh, for the principal glass works of the Macbeth-Evans Glass Company, in the year 1895. This establishment gives employment to many hundreds of persons. For several years prior to his death he was unable to give this ably managed establishment much attention. He never, however, forgot the sick and unfortunate among the deserving employees, and in Charleroi, the compiler is informed, are many who were called upon to mourn the loss of a beneficent employer.

In 1880 he married Miss Katherine Duff of this city. Mrs. Macbeth died 1911. Two daughters—Mrs. Anna von Moschzisker, wife of Justice Robert von Moschzisker of the Pennsylvania Supreme Court, and Mrs. Helen Whitehall Boggess, wife of Dr. W. B. Boggess of this city—and one son, George Duff Macbeth, who was associated with his father in the Macbeth-Evans Glass Company, survive.

Robert Sherrard Orr

Limestone, Clarion Co., Pa., October 14, 1867.
Pittsburgh, January 24, 1917.
Joined the Society, October 1899.

Robert Sherrard Orr died in St. Francis Hospital the afternoon of January 24, 1917, after a brief illness caused by an affection of the tonsils. At the time of his death he occupied the office of Vice President and General Manager of The Duquesne Light Company, Pittsburgh.

Robert Sherrard Orr was born on a farm near Limestone, Clarion County, Pa., October 14, 1867. His parents were the late Culbertson and Susan Sherrard Orr. He was educated in the public schools of Clarion County, and at the solicitation of his aunt, Miss Nancy Sherrard, principal of Washington Female Seminary, and famous as an educator, he was sent to Washington and Jefferson Academy, and later entered Washington and Jefferson College. He was graduated with honors in 1891, with the degree of Bachelor of Arts. He accepted a position as Instructor in Washington and Jefferson Academy, leaving there the following year to become Principal of the Ninth Ward Public School of Allegheny, a position he occupied for twelve years. He became well known as an educator, taking a leading part in educational affairs throughout the State, as well as in his own community. He made many improvements in the school under his management. He introduced manual training for the boys and domestic science for the girls, being a pioneer in these important branches of modern education. During this time he made a special study of electricity, becoming so proficient that he was selected as General Contracting Agent of the Allegheny County Light Company in 1904. His great ability, honesty, industry and fairness won him recognition and promotion, so that he became the General Superintendent, General Manager, and later the Vice President of the Duquesne Light Company.

High honors came to him in the electrical profession. He was active and prominent in state and national electrical organizations.

Too much cannot be said of his fine, manly character. He held the respect and admiration of all men who worked with him, in any capacity whatsoever. There was in him such a rare combination of intellectual ability, executive power, sincerity, fair mindedness, big heartedness, modesty and gentleness, as made him the friend of all who knew him.

He was Past President of the Pennsylvania Electric Association, Vice President of the National Electric Light Association, a member of the American Institute of Electrical Engineers, of the Franklin Institute of Philadelphia, of the Engineers' Society of Western Pennsylvania, of the Chamber of Commerce, University Club of Pittsburgh and the Pittsburgh Athletic Association.

He was married in 1912 to Miss Beryl Riggs. His wife died September 13, 1914. He is survived by two sisters and three brothers: Mrs. M. S. Hill of Tacoma, Washington, Miss Jane Orr of Limestone, Pa.; Edward F. Orr of Curtis, Washington, William E. Orr of Tacoma, Washington, and Charles F. Orr of Limestone, Clarion County.

Joseph Ramsey, Jr.

Pittsburgh, April 17, 1850.
East Orange, N. J., July 7, 1916.
Honorary Member, February 1903.

Joseph Ramsey, Jr., was born in Pittsburgh, Pennsylvania, April 17, 1850, and was of Scotch-Irish ancestry. He was the son of Joseph and Mary Patterson Ramsey. One of the members of the Ramsey family left Scotland in the year 1609 and settled in the north of Ireland, and from him was descended William Ramsey, who came to America in 1730 and purchased a large tract of land in East Nottingham, Chester County, Pennsylvania, and whose son, Joseph L., the grandfather of Joseph Ramsey, Jr., married Rebecca Starr, a member of the well-known Quaker family of that name. Mary Patterson was descended from James Patterson, who left Scotland in the seventeenth century and settled

in Virginia. Her father, Major Nathaniel Patterson, was born near Harrodsburg, Kentucky, and came to Pittsburgh with his father about 1800. Major Patterson fought in the War of 1812, and was very prominent in civic affairs. He was a Civil Engineer by profession, and served for many years as Engineer of Allegheny County.

Joseph Ramsey, Jr., received his education in the Bedford public school of Birmingham, now a part of Pittsburgh, and at the Western University of Pennsylvania. He did not graduate, however, as Mr. M. J. Becker, Chief Engineer of the Pittsburgh, Cincinnati & St. Louis Railway Company, applied to the University faculty for an engineering student, and Mr. Ramsey was recommended, and left the University in April, 1869, to take service with Mr. Becker, thus following in the footsteps of his grandfather Patterson from whom he doubtless inherited the traits which distinguished his professional career. His first work was with the engineer corps completing the Cork Run Tunnel and then locating and building the Dresden extension of the Cincinnati and Muskingum Valley Railroad from Zanesville to Dresden, Ohio, and in a few months he was made Engineer in Charge of the work; later he became Assistant Engineer of the Cincinnati and Muskingum Valley Railroad Company, leaving in May, 1871, to become Chief Engineer of the Bell's Gap Railroad, a narrow-gage road projected from Bell's Mills, Pa., to the extensive coal fields lying north of the Pennsylvania Railroad. The preliminary location having been completed, he served as Assistant Engineer of the Sunbury & Lewistown Railroad from October, 1871, to April, 1872, returning then to Bell's Mills to take charge of the work of building and equipping the Bell's Gap Railroad, and served as its Chief Engineer and Superintendent until February 1, 1879, when he resigned to become Chief Engineer and Superintendent of the Pittsburgh, New Castle & Lake Erie Railroad, a narrow-gage road in operation from Allegheny to Zelienople. But left that Company in November, 1879, to become Chief Engineer and Superintendent of the Pittsburgh Southern Railroad, also a narrow-gage road, with which he remained until April 1, 1882, when he was elected Chief Engineer and General Manager of the Pittsburgh, Chartiers & Youghiogheny Railway Company, hold-

ing similar official positions with the Chartiers Block Coal Company. He left Pittsburgh in August, 1883, to take service with the Cincinnati, Hamilton & Dayton Railroad, as Engineer, becoming Chief Engineer in 1886. His work in this new field attracted the attention of Mr. M. E. Ingalls, President of the Cleveland, Cincinnati, Chicago & St. Louis Railway Company, commonly known as the "Big Four." On January 1, 1890, Mr. Ramsey was appointed Assistant to the President of that company, and on June 1, 1891, was elected General Manager in charge of maintenance and operation, which position he held until April 1, 1893, when he was elected General Manager of the Terminal Railroad Association of St. Louis, of which all the railroads entering St. Louis were members. During his service with the "Big Four," Mr. Ramsey was President or Vice President of a number of the subsidiary or allied Companies belonging to that system. His success in handling the affairs of the Terminal Association led to his being elected, on December 1, 1895, Vice President and General Manager of the Wabash Railroad, being elected President on June 20, 1901, which position he held until April 19, 1905. He was also President of the Wabash-Pittsburgh Terminal Railroad from May 9, 1901, to May 1, 1905, and President of the Wheeling and Lake Erie Railroad from 1903 to 1905, and from 1909 to 1915. Mr. Ramsey became connected with the Ann Arbor Railroad as Vice President in 1907 and served as its President from 1910 to 1912. In 1905 he became interested in railroad properties in Lorain and Ashland Counties, Ohio, and caused the Lorain and Ashland Railroad to be incorporated—of which he became President—to build from Ashland, Ohio, to Lorain, Ohio; this road and the Industrial Railroad at Lorain, Ohio, being consolidated in November, 1910, as the Lorain, Ashland and Southern Railroad, which in 1913 absorbed the Ashland and Western Railroad, Mr. Ramsey being made its President, which position he held at the date of his death. After severing his connection with the Gould system, he was engaged by Mr. Harriman to locate a low-grade railroad from Chicago to New York. He was engaged also in the management of several syndicates with which he had been connected while with the Wabash Railroad.

Mr. Ramsey in his early professional life was an ardent advocate of the narrow-gage railroad, writing many interesting articles on the advantages thereof for the Railroad Gazette. He was perhaps at his best as a locating and constructing engineer, and his most prominent engineering work was the Wabash-Pittsburgh Terminal Railroad built to connect the Gould System with Pittsburgh, although, owing to the breaking down of the plan for a great transcontinental Gould System, the Pittsburgh enterprise proved a financial failure.

Mr. Ramsey was married April 8, 1873, to Laura Palmer, who, with four children, survives him. He was a member of the Evangelical Lutheran Church, and was elected a member of the American Society of Civil Engineers May 1, 1889. He was also a member of the Engineers' Society of Western Pennsylvania. He possessed a scientific mind, was an original thinker, and an indefatigable worker. His only recreation was chess, and the last evening of his life was spent over his chess board. He was devoted to his family and kindred, kind and considerate to his subordinates, and loyal to his friends, and in his death the members of this Society have lost a valued friend and associate.

George W. Schluederberg

Philadelphia.
Pittsburgh, April 1, 1917.
Joined the Society, March 1892.

George W. Schluederberg was born in Philadelphia 74 years ago, and at the age of two years removed to Pittsburgh, where he afterwards resided. At the time of his death, April 1, 1917, he was General Manager of the mines of the Pittsburgh Coal Company. He had been ill but five days with pneumonia, and until stricken had led a very active life and filled an important place in Pittsburgh's industrial field.

Mr. Schluederberg was one of the pioneers in the coal mining industry in this country, and probably no man was better ac-

quainted with the industry in connection with the central producing states than he. He was closely affiliated with every phase of the Joint Inter-State Scale movement, which embraced the four competitive districts of Western Pennsylvania, Ohio, Indiana, and Illinois, and had been for many years a member of the Scale Committee which negotiated wage scales for these districts.

He had traveled extensively, making many visits to different coal properties in Pennsylvania, Ohio, Indiana, Illinois, Kentucky, Tennessee, and West Virginia, and recently made a trip to Japan for the Pittsburgh Coal Company and for recreation. He was an indefatigable worker, and on a number of occasions, in connection with mining disasters, demonstrated his great ability and vitality.

Mr. Schluederberg joined the Engineers' Society of Western Pennsylvania in March 1892, and since that time had been a constant member, attending as many of the meetings as his busy life would permit. He was a patron of arts and music and also a historian of repute, being a member of the Historical Society of Western Pennsylvania. He was also a member of the German Evangelical Church, and of Solomon Lodge, F. & A. M. He was a Past President of the German Club, and was also a member of the Pittsburgh Art Society.

Robert Swan

Allegheny, Pa., October 18, 1859.
Pittsburgh, October 14, 1916.
Joined the Society, February 1883.

Robert Swan was born in the old City of Allegheny on the eighteenth day of October 1859, a son of John Swan, a prominent contractor and at one time postmaster of that city.

After the customary attendance at the public schools, he entered the Western University of Pennsylvania and graduated from the Engineering Department of that institution in 1877.

On October 12, 1886, Mr. Swan was married to Miss Georgia Clark, and at his death left surviving him his widow, six sons and one daughter.

In 1882 he was appointed Assistant Chief Engineer of the Baltimore & Ohio Railroad Company, and while in this service was placed in charge of important work on tunnel construction near Mount Royal Station, north of the city of Baltimore. Being offered the responsible position of Engineer of his native city of Allegheny, he accepted and held this office for a number of years with great credit to himself and to the entire satisfaction of all concerned. He resigned to become Vice President and General Manager of T. A. Gillespie Company of Pittsburgh, and as such, was actively employed on work of such magnitude as the Hudson River Tunnel at West Point, the Erie Canal and the great Pittsburgh filtration plant at Aspinwall.

From his earliest labors till the close of his career he displayed the same sterling qualities of head and heart, sound judgment, courage, firmness, unflagging industry and high power of concentration; with genuine sympathy, kindliness, sincere cordiality, and a helping hand with a winning smile, for all who came within the charm of his presence.

His was a most lovable character. Forceful and firm in his convictions, he was never obstinate or unreasonable and was ever ready to give kindly consideration to the judgment or opinions of others. Eminent success in his chosen profession had not touched his native modesty, and he was willing always to bestow praise on others rather than seek it for himself. His great trait of character, his dominant thought on every case and occasion was to be fair and just to all parties. Technicalities were brushed aside, fine points of distinction or advantage were discarded, and his mind went straight to the essential right of any question to be met and solved.

Mr. Swan, reared in this workaday atmosphere, bred to toil and effort, was, first of all, a Pittsburgh man, deeply imbued with a sense of his City's historic past, its present commanding position in the eyes of the world and its glorious promise for the years yet to come. By experience, education and training he was splendidly equipped for the duties of the office to which he was called, by a wise and happy choice, and which he accepted at a personal sacrifice. Broad visioned, yet never visionary, he planned and builded not for the present only, but with wise appreciation

of the future necessities and requirements of a great and growing city. With a comprehensive grasp of affairs, fortified by actual knowledge and skill acquired in the same field of endeavor, he took over the Public Works Department as a master of every detail. Each bureau and division felt the impulse of his militant spirit and responded to the supervision of his watchful eye; and best of all he infused into every employee his own pride in his work and the joys of achievement, as well as a deep respect and affection for himself. His service to the city and its people is appreciated, but not to the full limit of his deserts. Time must disclose and develop the splendid contribution to Pittsburgh's welfare made by Mr. Swan in the brief period of his official term.

His life is ended and those who knew his high purpose, his noble character, his faithfulness and his devotion to duty, will truly mourn him, but the good he has done will live after him as a fragrant memory and an enduring monument to perpetuate his name in the city he loved so well.

The esteem and honor in which Mr. Swan was held by his fellow citizens is well expressed by the resolution of City Council from which the foregoing is taken. In addition to the resolution of Council, many expressions of his ability and character were made by prominent citizens at the special meeting called by Council on October 27, 1916, to pay tribute to his memory. Mr. Swan was a member of the Engineers' Society of Western Pennsylvania, the Engineers' Club of New York, the Duquesne Club, and other social organizations.

Edmund Yardley

Yardleyville, Bucks County, Pa., September 9, 1836.
Pittsburgh, February 18, 1918.
Secretary of the Society 1907.
Joined the Society, October 1898.

Edmund Yardley who passed away on the eighteenth of February, 1918, was the son of Courtland and Hannah Ann (Brown) Yardley. He was born at Yardleyville, Bucks County, Pa., September 9, 1836.

His preparatory professional education was obtained at Classical Institute, Albany, N. Y. He edited and published a little paper at Albany in 1853, called "The Keepsake." That year he entered the Rensselaer Polytechnic Institute at Troy, N. Y., graduating in the Class of 1856, as Civil Engineer and at once entered upon the practice of his profession.

His first position was as rodman on the North Indiana Air Line Railroad, Ligonier, Ind.

From May to September, 1857, he was Principal Assistant Engineer of the Cincinnati & Mackinaw Railroad, going in April. 1859, to the Pennsylvania Railroad, in a similar capacity, on the Pittsburgh Division.

From 1864 to 1868 we find him on the Philadelphia and Erie Railroad (that road being leased by the Pennsylvania) as Resident Engineer, West Division, and located at Erie, Pa.

Later, 1868-1870 he occupied a similar position on the Pennsylvania, Pittsburgh Division, and was stationed at Pittsburgh.

In 1870 was moved to Altoona, Pa., where for the following three years he was in charge of construction of the new shops there.

Mr. Yardley then followed mercantile pursuits for the next six years, but returned to his first love in 1880 by being associated with Pool Commissioner Albert Fink. Later he returned to the Pennsylvania Company as General Car Accountant and Superintendent of Transportation, a position he held for many years until retired on account of age.

Mr. Yardley was a member of the American Society of Civil Engineers for some time and joined our Society in 1899, becoming Secretary for 1907.

During the construction of the large shops of the Pennsylvania Railroad at Altoona in 1871, the writer had the pleasure of being associated with Mr. Yardley for a short time, and looks back upon that experience with much satisfaction.

While at Erie, Mr. Yardley was in general charge of the construction of the machine shops there and also at Kane on the Pennsylvania Railroad.

At Altoona, besides taking care of the local work already referred to, he had general charge of the engineering along the

line between that point and Pittsburgh, such as laying out passenger stations and designing and constructing water stations, etc.

While a member of the American Society of Civil Engineers he contributed an article on Cements in 1872.

Mr. Yardley was of a genial disposition and a good attendant at the annual reunions of the Pittsburgh Alumni Association of R. P. I. He will be sadly missed at our future gatherings.

WILL THE PROPOSED LAKE ERIE AND OHIO RIVER CANAL IF BUILT BE AN ECONOMIC SUCCESS?

By W. G. Wilkins*

The writer first became acquainted with the fact that it was proposed to build a canal between Lake Erie and the Ohio River, in 1889, when the Governor of Pennsylvania appointed a commission to investigate and report on the subject, but did not pay much attention to the project until 1895, when the Pittsburgh Chamber of Commerce appointed a "Provisional Committee" to investigate and report on the "practicability and value to commerce by a ship canal connecting Lake Erie and the Ohio River via the Beaver and Mahoning Rivers." Since then he has read everything that could be found in print regarding the project, in the way of reports, acts of the legislatures of Pennsylvania, Ohio, and West Virginia, as well as newspaper clippings relating to the subject, and the paper read by Mr. Stickney, the Consulting Engineer of the Canal Board before this Society on April 20, 1915 (Proceedings, v. 31, pp. 285-333).

Most of the reports contain figures as to the traffic in ore and coal between Lake Erie and the furnace and coal districts of the three states mentioned, as well as estimates as to how much of this traffic the canal would obtain if it should be built. These latter figures at first sight appear to be such a small proportion of the total that, to the casual reader, it would seem as if there could be no doubt that the canal would obtain a tonnage sufficient to pay at least operating expenses, as well as interest and sinking-fund charges on the cost, to say nothing of any profits.

The 1907 prospectus of the Lake Erie and Ohio River Ship Canal Company said: "It will be observed that in the third year of operation the canal is self sustaining, and between the fourth and fifth year pays a dividend on the capital stock of six per cent."

*The W. G. Wilkins Co., Engineers and Architects, Pittsburgh.

The writer in his address before the Pittsburgh Chamber of Commerce on February 20, 1917, gave facts and figures to show "Why the Lake Erie and Ohio River Canal *should not* be built."

Since delivering that address he has received a copy of the report of the Canal Board dated June 28, 1917. This report states that the total tonnage in 1914 between Lake Erie and the Ohio River was 308 261 633 tons, and then goes on to show that a traffic of 19 000 000 tons would net the proposed canal a profit of $1 030 000. The average reader would be very apt to think that as 19 000 000 tons is only a little over six per cent. of the above-mentioned tonnage the canal would easily obtain at least that small proportion of the total traffic.

The fact is, however, that in none of the reports and pamphlets published regarding the canal, does there seem to have been any careful study and analysis of the traffic between the Lake Erie ports and the coal and furnace districts, to determine just how much of the traffic the canal might reasonably be expected to obtain, if it should be built. The writer has made such a study, and he believes that after a consideration of what follows, any unbiased investigator will arrive at the same conclusion that he has, viz., that if the canal should be built it would not obtain enough traffic to pay the operating expenses and bond charges—notwithstanding the fact that it has been under investigation for a period of over twenty-eight years, at an expense of nearly three hundred thousand dollars—and that it would be an economic failure no matter whether it was built by private capital; or from proceeds of bonds issued by the counties of the three states as proposed; or by the United States Government.

It was said by the president of the Canal Board, in his address, "Does Pittsburgh need the Canal?", before the Chamber of Commerce on February 13, 1917:

> "The great project must not be used as a political football, but considered as an economic question only, and the people should have the information to enable them to vote intelligently on the subject."

The writer agrees with this statement, and it explains his purpose in undertaking to give reasons why it would be an economic failure, and why the counties supposed to be benefited by

the canal should not issue millions of bonds in aid of its construction.

IS THE PROPOSED CANAL FEASIBLE?

One of the arguments used by the advocates of the canal is that ten great waterway engineers have pronounced the canal feasible. Now, just what does that word feasible mean? Webster's Dictionary gives the following definition:

> "FEASIBLE:—Capable of being done, executed or effected; practicable."

According to the same authority "practicable" means "capable of being used." The writer, in view of these definitions, admits that the canal is feasible from the physical, but not the financial standpoint, for he does not deny that it can be executed if the money is furnished, and that if executed it will be "capable of being used".

One of the highest functions of an engineer, and his duty as well, is to advise his client, not only whether the project he is reporting on is capable of being executed and of being used, but also whether if used it will be a financial success. In what follows, therefore, the writer will confine his remarks to the economic features of the question rather than to the purely constructive features.

There is in the writer's mind no doubt that the ore and coal traffic between the Great Lakes and the coal and furnace districts of Ohio, West Virginia, and Pennsylvania, is large enough to make the proposed canal a financial success, provided this traffic could be obtained. The writer, however, believes it would get practically no coal traffic, and to predict just how much of the ore business could be taken from the railroads that now carry it, can be only a guess. The writer's reasons for this statement will be given further on. In his opinion, it was not so much the panic of 1907 that was the cause of the failure to finance the building of the canal from the proceeds of the sale of the Canal Company's stock and bonds, as it was the fact that financiers and investors could not be convinced that the preliminary estimates of

traffic on the canal could be fulfilled, or the estimated profits by any possibility realized. The writer also believes it was because the promoters, finding it impossible to raise the required funds through the purchase of the Canal Company's securities by investors, decided that if they could get bonds issued by the counties of Pennsylvania, Ohio, and West Virginia, there might be some prospect of raising the money to build the canal, as the counties would be obliged to pay the interest on the bonds, and also the amounts required for sinking funds, even if the receipts from the operation of the canal should not be sufficient to entirely reimburse them.

It has been said by the president of the Canal Board, "The counties must loan their credit and use of some money for a few years." Now let us see just what this statement means. If to construct the canal ready for operation should take five years, which is the time estimated in the 1907 prospectus, and bonds of \$65 000 000 were issued — \$3 000 000 the first year, \$9 000 000 the second, \$16 000 000 the third, \$20 000 000 the fourth, and \$17 000 000 the fifth year; which is in about the same proportion as proposed in the 1907 prospectus—and if the interest rate were six per cent., the total interest for the five years during construction—which the counties would pay in addition to the \$65 000 000 —would amount to \$9 340 000. Even if the interest rate on the bonds were only four per cent., the interest during the five years of construction would amount to \$6 240 000; so that the total amount the counties would be called upon to pay up to the time the canal was ready for operation would be \$71 240 000 to \$74 340 000, which is the real amount of credit and money the counties must provide during the period of construction—and no sinking-fund payments were required until after the canal should be in operation.

If the Warren and New Castle branches were to be built—which the writer believes would have to be done if the Canal Board expects Lawrence County, Pennsylvania, and Trumbull County, Ohio, to vote favorably on the bonds—the total amount of bonds required would be \$72 000 000. If six per cent. bonds were issued, the total cost, including bond interest during construction, would be \$82 320 000, and if the rate were four per

cent., the cost would be $78 880 000, to which he believes the sinking-fund requirement during construction should be added, which would make the total cost in round numbers eighty-two millions if four per cent. bonds were issued; which does not include the cost of equipment of the Lake Erie terminal.

HAVE ANY EMINENT CONSULTING ENGINEERS REPORTED ON THE FINANCIAL PROSPECTS OF THE CANAL?

The president of the Canal Board in his Chamber of Commerce address gave the names of ten "great waterway engineers who have thoroughly investigated the question of the feasibility of the canal, *including the water supply, which is the question to which they have all addressed themselves.*" (The italics in this and certain subsequent quotations are the author's.) He did not, however, state whether any of these ten engineers had been asked, or if asked, had made any report on the financial question as to whether the canal, if constructed, would pay even operating expenses, to say nothing of interest and sinking-fund charges on the bonds. The writer has searched through every report he could find relating to the canal, beginning with the "Report of the Provisional Committee of the Chamber of Commerce" in 1897, and has not been able to find in any of them a report from a consulting engineer, on this feature of the project.

In his search for information, however, he came across the opinion of one of the ten waterway engineers referred to by the president of the Canal Board. It was found in a pamphlet dated 1911, entitled "Reports of Army Engineers upon the Proposed Canals Connecting Lake Erie with the Ohio River and with Lake Michigan." These reports were transmitted by the Chief of Engineers, U. S. A., to Hon. T. E. Burton, Chairman of the National Waterways Commission, and one of them is a report on the proposed Lake Erie and Ohio River Canal, signed by Col. (now General) H. C. Newcomer, from which the following paragraph is copied:

"While it is thought that the saving in canal rates compared with existing and even somewhat reduced freight rates would be sufficient to justify the construction of the canal, if this were necessary in order to

get the low rates, it is believed that if the canal were built the rates of the railroad would be reduced to such a point that the saving in shipment by the canal, would, in many cases, fail to warrant the shippers in giving up rail transportation. *It is very doubtful, therefore, in my opinion whether there would be sufficient commerce on the canal to provide a revenue commensurate with the expense of the work.* If the canal were free of tolls it would carry much more commerce and would save shippers considerable, just as it would save them considerable money if the railroad lines with their rights-of-way, were provided for their use without any charge upon their traffic."

This statement was made regarding a canal estimated to cost $52 000 000, and the financial success of a $65 000 000 or a $72 000 000 canal would be more doubtful, and that of one costing $82 000 000 more doubtful still. Colonel Newcomer also said regarding coal shipments by the canal: "It is understood that most of this [coal] now goes to the lake from mines that are not on a navigable waterway, and *it is doubtful whether rail shipments would be diverted to the waterway under such circumstances.*"

In the 1917 report of the Canal Board there is under the heading, "The Canal's Profits" an entire page which is intended to answer the question—"Will the canal yield proper returns on the money invested?" in which the Board gives an opinion that it will be indirectly profitable through decrease in transportation costs; that eventually the advantages of the canal will be recognized and it will carry a large tonnage; that the size and capacity of the canal are in keeping with the growing needs of the industries; that it will bring new industries to its immediate territory and finally that: "In view of these facts it is the belief of the Board that after a reasonable time, the traffic carried by this canal will bring a revenue sufficient to pay the cost and eventually retire the bonds."

A careful reading of the report fails to show that any opinion as to financial prospects, which would be received with confidence by investors, has been secured from a single waterway engineer of national reputation. The report is also silent as to whether the waterway experts employed by the present board have reported, or were even asked to report, on the financial features of the project.

The only one of the ten "great waterway engineers", as already shown, who has publicly expressed his opinion on this phase of the project, is Colonel Newcomer and he practically said that it is doubtful whether the canal would pay if it were to be built.

DOES THE FINAL REPORT OF THE NATIONAL WATERWAYS COMMISSION FAVOR THE CANAL?

It has been said that the National Waterways Commission recommended that the United States Government should construct the harbor at the Lake Erie terminus; recommended the deepening and improving of the Ohio River between Pittsburgh and the mouth of the Beaver River to allow 12-foot barges to be used, and recommended also the assigning of United States Engineer Officers to supervise the work. No mention, however, has been made of the strings attached to these recommendations in the way of conditions under which the Government was advised to do these three things. The conditions are:

1. Before the United States Government Engineers are assigned to the work the Canal Board *must have available* \$10 000 000 in cash. According to the Pennsylvania Canal Act no work can be started nor will any money be available until the State Treasurer has on deposit with him the entire estimated cost of the canal, either in money or in bonds at their par value.

2. When bonds to the amount of \$50 000 000, or as much more as may be necessary in the opinion of the Secretary of War to insure its completion, have been authorized, and the legality of such bonds been certified by competent legal authority, he shall . . . direct the Chief of Engineers to detail without charge such officers as he shall deem necessary to perform the engineering work required in the construction of the proposed canal.

3. The Commission further recommends but does not guarantee that when the work of construction is actually begun, Congress, *if satisfied that it will be completed,* shall appropriate the funds for an adequate harbor on Indian Creek at the Lake Erie end, and for the necessary improvement of the Ohio River in the Pittsburgh district; the same to be completed by the time the canal shall be ready for operation, but the Government shall not be required to purchase any land required in making these improvements.

It will be seen from the above quotations from the Report of the National Waterways Commission that there are some pretty stiff conditions attached to the recommendation for assistance from the United States Government, and the recommendation concludes with the following paragraph:

"The plan of co-operation here proposed is intended to leave the Canal essentially a local enterprise, and the above recommendations for the co-operation of the Federal Government in constructing this waterway are *not to be construed as committing or obligating the Government now or at any future time to assume any financial responsibility whatever in any way related to its construction, maintenance, or operation.*"

The Commission was composed of seven members of the United States Senate and five members of the House of Representatives, the chairman being former Senator Theodore E. Burton. On the first page of the report is found the following:

"This report presents the unanimous conclusions and recommendations of all the members of the commission on various questions relating to waterways, on the following subjects:

"1. The advisability of the Federal Government constructing the proposed Canal connecting Lake Erie with the Ohio River near Pittsburgh, the expense of which is to be borne by the local interests affected."

The following paragraph from the report, with regard to the amount of traffic the canal might obtain, does not appear to be so greatly in favor of the canal as to convince an impartial investigator that the members of the Commission were personally convinced that if built it would be financially successful:

"The main question to be considered in reaching a conclusion as to the feasibility of the proposed canal is what part of this traffic it can reasonably be expected to obtain in competition with the railways now operating in the same territory. The calculations made in 1905 give 3,000,000 tons as the probable traffic for the first year of operation, 22,500,000 for the fifth year, and 38,000,000 for the tenth year. *There are a number of considerations which would indicate that these estimates are probably too high.* In order to successfully compete with the railways the canal must offer much cheaper transportation, except when there is an excess of traffic. This is due to the fact, as set forth in the preliminary report of the commission, that water transportation has certain natural disadvantages which lessen its convenience and reliability, and cause shippers to patronize the railway in preference, unless the water route offers sufficient inducement in cheaper transportation."

The report of the Commission also included the following paragraph with reference to possible coal traffic on the canal:

"*It is the opinion of the Commission that the amount of coal which it is expected the proposed canal would carry has been overestimated. The coal deposits in the Monongahela Basin near to the river are becoming depleted so that it will be more expensive in the future to bring to the river the coal that is to be shipped in barges. This would make it more economical, as well as convenient, to patronize a railroad whose tracks reach directly to the mines.* Furthermore, some of the coal lands most favorably situated for shipment of their output by water are owned or controlled by persons or corporations affiliated with the railroad interests. Some allowance should also be made in the estimates of both prospective iron ore and coal traffic of the canal, from the fact that many of the large iron and steel industries using these raw materials are more or less affiliated with the railroad interests, which are the largest purchasers of their manufactured products."

Does not a consideration of the foregoing extracts from the Commission's report suggest the thought that so far as the Pittsburgh and Lake Erie Canal is concerned, it is a case of "damning with faint praise"?

TERMINAL FACILITIES

The terminal facilities that must be provided for the operation of the proposed canal would be:

1. The terminal harbor on Lake Erie with its wharves or piers on which would be placed the machinery for transferring coal from canal-boats to lake steamers, and ore from lake steamers to canal-boats.

2. The machinery for taking the ore out of canal-boats at the blast-furnace plants and putting it in the stock houses or stock piles; also such changes in the plant as are made necessary by the use of the canal for its ore delivery.

The first is a comparatively simple proposition so far as the design and construction of the mechanical plant are concerned, as machines similar to some of the ore- and coal-handling appliances already in use on the Lake Erie coal and ore docks, could be used with some modification. The most difficult factor of the problem to decide on in advance, as will be shown later, is the size of the

harbor and the number of coal- and ore-handling machines that would ultimately be required.

When it is considered that with few exceptions the blast-furnace plants on the Ohio, Monongahela, and Allegheny Rivers, have their stock houses between the furnace and the railroads now delivering their ore, it can be seen that the problem of how to hoist the ore from the canal-boats and get it around back of the furnace into the stock house, is much more difficult than the first problem. The solution of this second problem would be different for nearly every furnace plant, and the construction cost, interest, and operating cost would probably vary at the different furnaces.

The writer wonders if either the members of the Canal Board or their engineers fully realize the size and cost of the mechanical plant that would be required to handle the coal and ore tonnage at the Lake Erie terminal—to say nothing of that required at the furnaces to unload and store the ore—as well as the cost of constructing and operating the tipples on the Monongahela and the Ohio to unload the coal from the railroad cars into the canal barges. Some idea of what this means can be had when it it is known that at Ashtabula in 1911 a steamer loaded with 10 234 tons of ore was unloaded in four hours and one minute, or at the rate of 2555 tons per hour. As showing what is possible with ore-handling plants it may be stated that at Conneaut Harbor in 1912 a boat load of 10 636 tons of ore was unloaded in two hours and fifty minutes, or at the rate of 3758 tons per hour, and in 1915 at Cleveland 11 270 tons of ore were unloaded in three hours and forty minutes, or 3078 tons per hour. Coal has been unloaded from railroad cars into lake steamers at the different Lake Erie ports in 1916 at the rate of from 32 to 35 cars per hour, or from 1640 to 2294 tons—an average of 1962 tons per hour, or at the rate of from 27 to 38 tons per minute—the loads of the cars ranging from 39 to 56 tons. The capacity of the lake boats into which this coal was loaded ranged from 10 000 to 12 650 tons.

For a capacity of 38 000 000 tons in 240 days per year the canal would have to handle an average daily tonnage of 158 333 tons, and for the most efficient and economical operation it would mean one-half this amount, or 79 166 tons every day in each di-

rection. Even if the canal should be dug to a 12-foot depth the boats for these rivers could be loaded only for a 9-foot depth, until such time as the locks and dams in the three Pittsburgh rivers were increased from 9 to 12 feet. According to Mr. Stickney, if the canal locks are 56 by 400 feet, the ore and coal barges recommended by him could be loaded with only 2775 tons for a 9-foot depth, which would mean 29 boats each way daily. It would also mean that at least eight 10 000-ton lake steam barges, or 10 of 8000 tons capacity, loaded with ore, would have to be unloaded and reloaded with coal every day. If the operation of the canal worked with clock-like regularity, with 29 canal-boats arriving daily at the Lake Erie terminal, loaded with coal, and the same number leaving with ore, the size of the terminal harbor and its equipment for loading ore and unloading coal would have to be adequate for not less than that number of boats. If it should happen, as in the case of the old Erie Canal, that the open season should be only 204 days, it would mean an increase of nearly 20 per cent. in the total boat and terminal capacity to maintain the 38 000 000-ton capacity.

Since the above was written the writer learns from the report of the Canal Commissioners, that they have decided that, instead of the power barges recommended by their consulting engineer in 1915, "the most economical carriers that can be used will be barges of at least 1100 tons capacity each, towed in fleets of three, by tugs or towboats." This would mean, instead of 29 trips of power barges daily, 24 trips of a towboat with three barges each way daily, and two and one-half times as much shifting of barges as with those of 2775 tons capacity.

It is not probable that the canal could be run with such regularity; so not only the size of the terminal harbor must be larger than for the above named number of boats, but also the loading and unloading plants must be larger, though just how much larger can not be foretold. The writer also believes, so far as the lake terminal is concerned, that it is impossible to estimate correctly the amount which should be added to the estimate to cover the cost of the terminal equipment.

COST OF CANAL IMPOSSIBLE TO PREDICT

In the report of the Canal Board, filed with the Governor of Pennsylvania, June 28, 1917, is an itemized estimate of the cost of the canal (Table I), and there is also a statement that the costs of six different types of canal were considered, but that the cost of the one recommended by the Board would be:

"For the main line of the canal with the necessary feeders and reservoir, about $65,000,000.

"In addition the cost of the branch to New Castle will be about $3,500,000 and the branch to Warren about the same. *These branches* will be valuable adjuncts to the canal and *should be built at the same time.*"

These two paragraphs certainly mean that the total cost of the canal as recommended by the Commission would be at least $72 000 000, to which should be added interest on the bonds during construction, and also—the writer believes, as a matter of sound financial principles—sinking-fund charges.

The Board, on the same page, asks the question, "What will the canal cost?" and then continues in the following words:

"The estimates of cost which follow were made by competent engineers of high standing. They have been checked and rechecked several times by other competent engineers. They probably are as nearly exact as any such estimates can be made. They were made in 1914-15 before prices had been increased by the European War and so must be revised if the canal is built before prices again become normal. *They do not include interest accruing on bonds during the period of construction.* Such interest, if any, can only be estimated when the method of financing and the terms and conditions of the bonds are known, and these matters will not be finally decided until the question of building the canal is submitted to the voters. A rough calculation might perhaps be made, but the assumptions involved on such a calculation would necessarily be so numerous that it would be meaningless."

It seems to the writer that the Board has answered its own question in this last paragraph, for the statements made in it can mean only, "We don't know what the canal will cost;" and the writer would add that in his opinion no one can come anywhere near predicting what the actual cost will be, if it is ever built; and he does not hesitate to make this prediction in view of the

history of the great canals previously built and now in operation.

TABLE I

ESTIMATED COST

Canal, Mouth Beaver River to Indian Creek (101.5 Miles).

	Quantity	Amount
Land	12 380 Acres	$ 1 995 000
Damages		2 950 000
Excavation, Earth	29 479 800 Cu. Yds.	6 725 000
Rock	4 230 000 Cu. Yds.	9 600 000
Embankment	3 572 600 Cu. Yds.	540 000
Slope Paving	450 000 Sq. Yds.	564 000
Dams, Fixed	1	
Bear Trap	2	
Bridge	9	1 880 000
Locks	26	14 300 000
Guard Gates	3	125 000
Spillways, Culverts, Aqueduct		1 200 000
Retaining Walls		1 050 000
Terminals	16	2 650 000
Bridges, Railroad	30	2 640 000
Highway	59	1 350 000
Total		$47 569 000

WATER SUPPLY

French Creek Feeder	$ 3 170 000
Mill Creek Feeder	1 250 000
French Creek, Reservoir A	950 000
Land, Reservoir AA (Cussewago) and Additional Land for Bemus Dam	245 000
Pymatuning Reservoir F, if Constructed for Canal Needs Only	2 862 000
Mill Creek Reservoir I	150 000
Total	$ 8 627 000

Canal, Mouth Beaver to Indian Creek	$47 569 000
Water Supply	8 627 000
	$56 196 000
Engineering and Contingencies	8 429 400
Total cost	$64 625 400

In this estimate of cost there are the two following items:

Bridges,	Railroad	30	$2 640 000
"	Highway	59	1 350 000
		89	$3 990 000

Mr. Stickney, in his above-mentioned paper before this Society, said that the Beaver and Mahoning Rivers and Mosquito Creek are crossed by 42 railroad bridges and 38 highway bridges, most of which will require modification to permit the canal-boats to pass under them. In addition, he said there would be from 39 to 51 new bridges, according to the route selected, or a total of from 119 to 131 bridges to carry railroads and highways over the canal, or from 30 to 42 more than given in this 1917 estimate.

The 1915 Pennsylvania Canal Act contains this proviso:

"Where said canal, its branches or channels, shall cross any existing public highway at a point where no waterway existed before, the Board at its own expense shall cause to be *constructed* and *maintained* suitable bridges to provide for traffic existing at the time of such *construction*.

"The Board in lieu of maintaining such bridge may pay to the proper authorities having jurisdiction over the highway, or to the owner of the railroad, as the case may be, a sufficient sum of money to provide for their perpetual maintenance."

It will be seen that the estimate includes $3 990 000 for railroad and highway bridges, but it does not seem to contain any amount for modifying the 42 existing railroad bridges and the 38 highway bridges, nor any item required by the Pennsylvania Canal Act for "a sufficient sum of money to provide for the perpetual maintenance of the new bridges." Was this proviso in the Canal Act overlooked in making the estimate?

There is another factor that might enter into the cost of the canal, which would make it impossible to predict the actual cost, provided the work were done under the supervision of the United States Engineers, and which the report of the National Waterways Commission says was proposed when they went over the route of the canal in 1911. In a letter from Gen. W. H. Bixby, dated June 10, 1911, to Senator Burton, the Chairman of the Waterways Commission, he says:

"While it is believed desirable and proper that the work should be planned and executed under the supervision of the Engineer Corps, the details of the legislation providing for such an arrangement would require careful consideration, for if the funds furnished by the State or local agency for the execution of the work were deposited in the Treasury of the United States, as is the usual case, and the work executed as an improvement by the United States, it would be subjected to the restrictions of the eight-hour law, civil service and department regulations, *which would materially increase the cost of the project.*"

There is also another item which does not appear to be included in the estimate but which will probably be part of the cost of the canal if it should be built. This is in connection with the storage reservoirs which it is proposed to build on the head waters of French Creek for the purpose of providing additional water storage to be used when the water in the Pymatuning Swamp reservoir became low in time of drought, which would be just the time when the farmers in the valley below these reservoirs would be needing water. With regard to taking water from French Creek and from the drainage area of Pymatuning Swamp to furnish water for the purpose of navigation in the Beaver and Mahoning Rivers, the following principles laid down by a Subcommittee on Dams and Water Power, of the Committee on Interstate and Freight Commerce of the House of Representatives in 1909, seem to be pertinent:

"The United States possesses no right to the use of the water flowing in non-navigable streams and its use for navigation purposes, whether returned to the same stream *or diverted to another, must be paid for.* The use of such flowing water for navigation purposes in *an artificial waterway* may directly injure the property rights of state or individual in water powers, drainage, sanitation or consumption for domestic purposes."

If the United States cannot do this without compensation, the Lake Erie Canal Commission cannot do it, and the writer would ask if, in the estimate of cost of the reservoirs, any amount has been included to pay for the water, taken from the drainage areas of the streams mentioned above, which is to be used for navigation purposes in the proposed canal—an artificial waterway.

Mr. Stickney in his paper before before this Society said "At the Lake Erie end of the canal a *large terminal* with facilities for transferring freight from lake vessels to canal boats will be

necessary" and the chairman of the Canal Board in his address before the Pittsburgh Chamber of Commerce on February 13, 1917, said "The National Waterways Commission recommended that the United States Government construct a *harbor* at Indian Creek where the canal empties into Lake Erie".

The recommendation of the Waterways Commission reads that Congress "If satisfied that it [the canal] will be completed, shall appropriate the funds necessary for an *adequate harbor* in Indian Creek at the Lake Erie end".

It will be noticed that while Mr. Stickney says a "large terminal" will be necessary for transferring freight from the lake vessels to the canal-boats, the Waterways Commission recommends that the Government construct simply a *harbor* at Indian Creek and says nothing about the wharves or piers at which the boats would be loaded and unloaded, or the mechanical equipment required to make the lake terminal complete.

The estimated cost given in the report includes $2 650 000 for 16 terminals, but the writer has been informed that the cost of the wharves and equipment at the lake terminal is not included in that sum. If this is the case, and the Government should simply dredge out the harbor it would mean a further addition to the estimate to cover the cost of the wharves and the mechanical equipment for the Lake Erie terminal, but, as already shown, it is impossible to predict just what would be the final cost of the complete terminal, and it follows therefore that it is impossible to estimate in advance what the actual cost of the canal would be, complete and ready for operation. In connection with the subject of cost, attention is called to the fact that the plans of the Canal Board are to build a single- instead of a double-lock canal. According to the authority of Col. Thomas P. Roberts, who has had many years experience with the locks on the Monongahela River: "With all possible care, accidents to lock gates or valves are apt to throw a lock out of commission for days or even weeks at a time if duplicate locks are not provided." In view of these facts, does it not mean that in order to keep down first cost and to commit the public to the project, the Canal Board is willing to build a canal which, according to Colonel Roberts is liable to have the traffic interrupted for days or weeks at a time?

WHY HAS THE WIDTH OF THE CANAL BEEN CHANGED FROM 180 TO 140 FEET?

The June 1917 report of the Canal Board says:

"In answer to the question what should be the dimensions of the canal and locks, the Board is of the opinion that the canal should be built with a *bottom width of 140 feet,* a depth of 12 feet and single locks 56 x 400 feet, 12 feet over the sills".

Mr. Stickney, in his paper before this Society said that "If the locks were 45 feet by 340 feet the canal need be only 140 feet in width *but if they were built 56 feet by 400 feet the bottom width of the canal should be 180 feet";* and also that "The type of boat best adapted for the transportation of bulk freight and package freight on the canal is believed to be power driven barges of the largest dimensions that the depth of water and size at locks will accommodate and which requires a minimum height of bridges". Mr. Stickney also wrote the Board, May 12, 1917, a communication in which he said:

"While a channel 140 ft. wide might be navigated with boats of the larger size, the width is not sufficient for economical operation and the cost per ton would be increased. The proper width of channel for the larger boats is not less than 180 feet".

Why has the Board ignored Mr. Stickney's opinion as to the width of the canal and size of boats, both of which are of great importance in the practical operation of the canal if it should be built? Is it because Mr. Stickney, in January 1916, wrote the Board that the canal 180 feet wide would cost $89 437 230? Were the members of the Board doubtful of its being built if it were to cost that amount of money? It looks very much as if that might be the reason why the Board was of the opinion that the width should be 140 feet, when taken in connection with the estimate of $64 625 400 for the narrower width, or nearly $25 000 000 less than for the canal 180 feet wide advised by their consulting engineer. This additional cost would mean $1 000 000 per annum for interest on 4 per cent. bonds, to say nothing of the additional sinking-fund charges, both of which would mean an increase in the canal toll charges over what would be necessary for the nar-

rower width canal. Then, too, is it not possible that the Board had an idea that it would be easier to secure a favorable vote on bond issues of $65 000 000 than on nearly $90 000 000? The tax-paying public could realize what the extra $25 000 000 in cost meant, but might not know that the Canal Board's consulting engineer had said it would cost the boat-owners more per ton for operating their boats in the narrower but cheaper canal. May not these have been some of the reasons the Board had for not accepting the opinion of its consulting engineer?

TABLE II

POPULATION AND VALUE OF PROPERTY IN THE CANAL ZONE

No.	County	Population 1910 Census	Assessed Value of Property
	Pennsylvania		
1	Lawrence	70 032	$ 33 779 152
2	Beaver	78 353	49 767 541
3	Allegheny	1 018 463	1 213 752 050
4	Washington	143 680	123 139 789
5	Greene	28 882	42 753 158
6	Fayette	167 449	94 169 416
7	Westmoreland	231 304	166 841 211
8	Armstrong	67 880	16 049 782
	Total	1 806 043	$1 740 252 099
	Ohio		
1	Ashtabula	59 547	$ 90 752 110
2	Trumbull	52 766	97 106 580
3	Mahoning	116 151	230 518 380
4	Columbiana	76 619	96 380 980
5	Jefferson	65 423	82 592 450
6	Belmont	76 856	*81 000 000
7	Monroe	24 244	21 533 275
8	Washington	45 422	47 243 000
9	Athens	47 798	*13 000 000
10	Meigs	25 594	18 249 790
11	Gallia	25 745	14 833 689
12	Lawrence	39 488	30 606 956
13	Scioto	48 463	56 519 530
	Total	704 116	$ 880 336 740

*Approximate.

WEST VIRGINIA

1	Hancock	10 465	$ 11 827 315
2	Brooke	11 098	16 244 751
3	Ohio	57 572	84 898 545
4	Marshall	32 388	40 138 029
5	Wetzel	23 855	34 256 146
6	Tyler	16 211	18 629 869
7	Pleasants	8 074	7 517 495
8	Wood	38 001	46 351 923
9	Jackson	20 956	9 213 729
10	Mason	23 019	13 012 572
11	Cabell	46 685	39 531 824
12	Wayne	24 081	17 093 485
13	Putnam	18 587	8 924 343
14	Kanawha	81 457	61 750 618
15	Monongalia	24 334	49 451 172
16	Marion	42 794	63 355 673
	Total	479 557	$ 522 197 289

SUMMARY

8	Pennsylvania	1 806 043	$1 740 262 099
13	Ohio	704 116	880 336 740
16	West Virginia	479 577	522 197 289
37	Counties	2 989 736	$3 142 786 128

WHEN WOULD IT BE POSSIBLE TO BEGIN AND COMPLETE CONSTRUCTION OF THE CANAL?

The Canal Commission has to date never furnished the public with any list of the counties of the three states—Pennsylvania. Ohio, and West Virginia—which are supposed to be directly interested in the construction of the canal, nor any statement as to the amount of bonds each county will be expected to issue in order to obtain the funds required.

The writer has, however, obtained a copy of a statement prepared by Mr. Stickney, Consulting Engineer of the Canal Commissioners, of the counties in Pennsylvania, Ohio, and West Virginia, with their assessed property valuations; which he suggests as the counties that would be interested enough in the proposed canal to vote bonds for its construction. This statement is given in full (Table II), and it will be seen that the total assessed val-

uation of these counties is $3 142 786 128, on which two per cent. would be $62 855 722, or a little less than the estimated cost of the canal, but not including the $3 500 000 required for each of the Warren and New Castle branches.

It will be seen that of the 37 counties in the list, 24 front on the Ohio River. Only two of these, Allegheny and Beaver, are in the state of Pennsylvania. On the Ohio side of the river the list includes all the 10 counties from Columbiana on the north, on down the river to and including Scioto, and on the West Virginia side all the 12 counties from Hancock down to Wayne, as well as one, Putnam County, in the New River district. On the Monongahela River, above Allegheny County, the list includes Washington, Greene, Fayette, and Westmoreland; and in West Virginia; Kanawha, Monongalia, and Marion. There are two other Pennsylvania counties in the list; Armstrong on the Allegheny River, and Lawrence, through which both the Beaver and Mahoning Rivers run. The list also includes Ashtabula, Trumbull, and Mahoning, the three Ohio counties through which the canal route is located. The writer has recently learned that another list of 53 counties has been prepared by the Commission, but has not seen a copy of it.

The 1915 Pennsylvania Canal Act contains the following proviso regarding the conditions under which work of construction can be started:

"*The work of constructing said canal or waterway and appurtenances shall not be proceeded with; nor any expenditures made, nor any contracts entered into, or any liability incurred therefor until there shall have been rendered available to the Treasurer of the Commonwealth, by the aforesaid public authorities* (i. e.,— the states and counties referred to above) or some of them in aid of the construction of such canal or waterway and appurtenances, contributions or appropriations in *money or bonds at their par value, at least equal in the aggregate to the cost of constructing, completing and installing such canal or waterway and appurtenances* as estimated by the board, of which estimated cost two-thirds shall have been authorized to be contributed by said counties or some of them."

The chairman of the Canal Commission has said, "By our present plan we can have the canal open in 1923." Now, in view of the foregoing is it probable or even possible that the pre-

diction regarding 1923 as the date of opening can be fulfilled? Even after the decision is made as to the counties to be asked to vote bonds, and the final apportionment of the estimated cost among the counties selected by the Commission as the ones to be benefited by the construction of the canal, petitions must then be secured asking for the elections in accordance with the canal acts of the three states, and the elections must be held. If the result in any of the counties in the three states should not be in favor of the bond issue, then the work cannot be started nor can any contracts be let. If this should be the result there must be a reapportionment of the sum estimated as the cost, and new elections must be held, or perhaps the Canal Commission might take a chance on asking one or more of the counties to vote the bonds required to make up the deficiency.

The writer personally does not believe that even all the 37 counties selected by Mr. Stickney as being benefited by the canal, to say nothing of the counties in the other list, will vote favorably on the bonds, and if such is the result it means that other counties must make up the deficit before the work of constructing the canal could be begun.

The date mentioned above for the opening of the canal is less than six years in the future and no movement has yet been made for the required bond elections. Even if the elections were held with results favoring the bonds, it would take at least five years for the construction of the canal. It therefore seems to the writer than even under the most favorable conditions it would not be possible to open the canal in 1923.

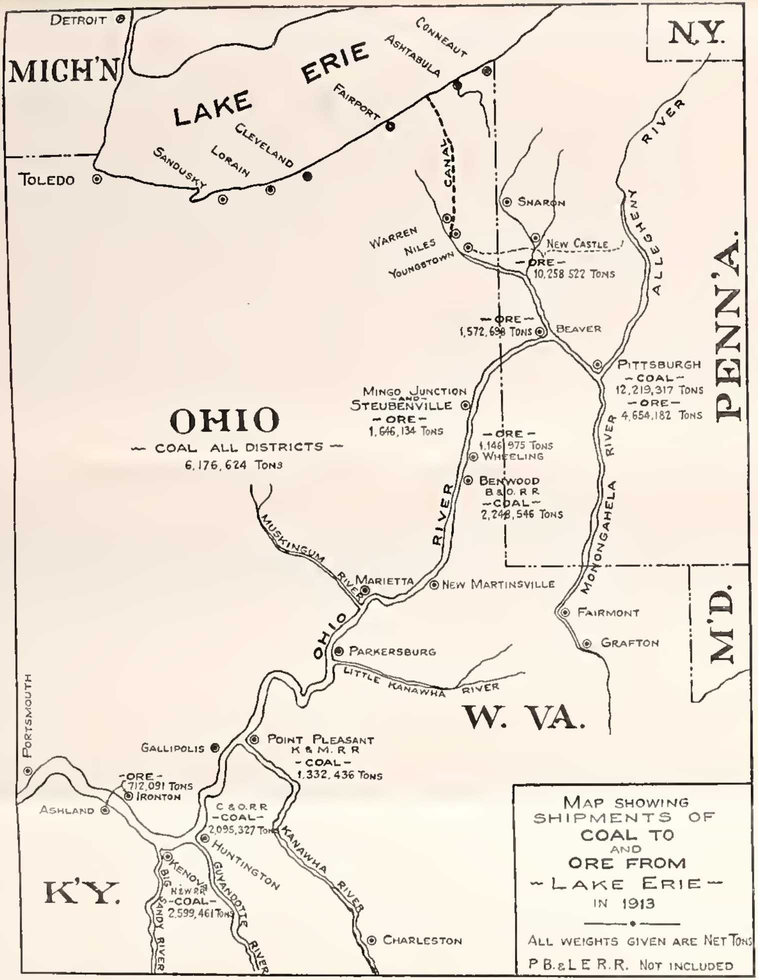
COST OF BARGE OPERATION IMPOSSIBLE TO PREDICT

Mr. Stickney said that from estimates of cost given by a marine architect who prepared plans of boats for carrying both bulk and package freight, it has been determined that for bulk-freight carriers, built for a nine-foot canal, the cost per ton mile for eight months continuous operation with full cargoes, will be 0.67 mills; for 12 feet 0.51 mills, and for 15 feet 0.43 mills, while for package-freight carriers the cost will be 2.89, 2.36 and 2.03 mills respectively for the three depths, for a distance of

150 miles from the lake terminal to points on the Ohio and Monongahela Rivers. These figures might be correct if every trip of a boat were made with a full cargo in both directions, but the writer believes these conditions could never be obtained in the proposed canal. He is of the opinion that it is impossible for any one to predict, before the canal is in operation, for what part of its round trip a boat will have a full load of either coal or ore, or for what portion of the trip the boat will be carrying no cargo at all. It is certain, however, that unless every boat carries a full load of ore to the same destination on every trip, and then goes empty to some regular loading place, and waits the same length of time for its return load of coal, the cost per ton-mile will exceed the above figures and will differ with every variation in the trips.

It is also certain, if the canal is ever built, that owing to the difference between the tonnage of ore to be shipped southbound and of coal to be shipped northbound—to and from the various furnace and coal districts—many of the boats will make their trips empty for some distance, either in one direction or the other, for at least part of every trip; and the writer believes that, owing to this fact, the cost for transporting coal from the mines to the Lake Erie terminal, or ore from the lake to the furnaces, would on many trips be greater than by rail. This is particularly true with regard to coal from the coal districts of Ohio and West Virginia, as will be shown later.

The writer has had a map prepared (Plate 1) showing the amount of ore shipped from Lake Erie ports to the different furnace districts in 1913, and also the amount of coal shipped to Lake Erie ports from the coal districts of Pennsylvania, Ohio and West Virginia during the same year. The year 1913 has been selected for the reason that up to date it is the year in which occurred the largest shipments of coal from the Pittsburgh district to Lake Erie ports—destined for the upper lake ports—and also for the reason that it is the latest year for which he could obtain the data on ore shipments. Table III is a tabulation showing in detail the ore and coal for each district and the totals of the shipments for that year.



Map showing
shipments of
coal to
and
ore from
~ Lake Erie ~
in 1913
All weights given are Net Tons
P.B.&L.E.R.R. Not included
Detroit
Mich'n
Lake Erie
Toledo
Sandusky
Lorain
Cleveland
Fairport
Ashtabula
Conneaut
N.Y.
Penn'a.
M'd.
W. Va.
K'y.
Ohio
~ Coal all districts ~
6,176,624 Tons
Canal
Sharon
New Castle
Warren
Niles
Youngstown
– Ore –
10,258 522 Tons
– Ore –
1,572,698 Tons
Beaver
Pittsburgh
– Coal –
12,219,317 Tons
– Ore –
4,654,182 Tons
Allegheny
River
Mingo Junction
and
Steubenville
– Ore –
1,646,134 Tons
– Ore –
1,146,975 Tons
Wheeling
Benwood
B.&O. R.R.
– Coal –
2,248,546 Tons
Monongahela River
Ohio River
Muskingum River
Marietta
New Martinsville
Fairmont
Grafton
Parkersburg
Little Kanawha River
Portsmouth
Gallipolis
Point Pleasant
K.&M. R.R.
– Coal –
1,332,436 Tons
– Ore –
712,091 Tons
Ironton
Ashland
C.&O.R.R.
– Coal –
2,095,327 Tons
Huntington
Kanawha River
Kenova
N.&W.R.R.
– Coal –
2,599,461 Tons
Big Sandy River
Guyandotte River
Charleston

TABLE III

Districts	Ore (Net tons)	Total	Coal (Net tons)	Total
Mahoning & Shenango Valley	10 258 522			
Beaver	1 572 698	11 831 220		
Pittsburgh	4 654 182	16 485 402	12 219 317*	
Mingo Junction & Steubenville	1 646 134	18 131 536		
Wheeling	1 146 975	19 278 511		
West Virginia—				
Benwood			2 248 546	14 467 863
Pt. Pleasant			1 332 436	15 800 299
Huntington			2 095 327	17 895 626
Kenova			2 599 461	20 495 087
Ohio			6 176 624	26 671 711
Ashland & Ironton	712 091	19 990 602		

This tabulation does not include the shipments by the Bessemer & Lake Erie Railroad, for the reason that the canal promoters say they do not expect to take away any of the business of that road. Neither does it include any of the coal which goes to the various lake ports and is used locally—because coal intended for local use at lake ports, either east or west of the terminus of the canal, would never be transported by the canal, unloaded from the canal-boats at the lake terminus and reshipped either by rail or water to its ultimate destination. A study of the tabulation will show that if the canal had been in operation in 1913, and ore boats coming to Pittsburgh with a capacity of 4 654 182 tons had been reloaded with coal, there would still have been needed additional boats of a capacity of 7 565 135 tons for transporting to the lake the remainder of the Pittsburgh district coal, the total of which amounted to 12 219 517 tons. These boats could have been obtained from those that carried the 11 831 220 tons of ore to the Mahoning, Shenango Valley, and Beaver districts, which would have left in those districts empty boats of a capacity of 4 266 085 tons. These boats, added to those (with a capacity of 2 793 109 tons) which carried the ore

*The shipments of lake coal from the Pittsburgh district have grown less each year, the figures for 1916 being 8 642 200 tons—a decrease of nearly 30 per cent. in three years.

to the Mingo Junction, Steubenville, and Wheeling districts, would have given empty boats of a total capacity of 7 059 194 tons, and they could all have gone to Benwood, Point Pleasant, Huntington and Kenova, for the West Virginia coal, which amounted to 8 275 770 tons, and which might have been brought by rail to these points. There would, however, still have been required for this coal, empty boats of a capacity of 1 216 576 tons. The boats that carried the ore to Ashland and Ironton, however, would have reduced the boats required to a capacity of 504 485 tons, which, with the boats that would have been needed for the 6 176 624 tons of Ohio coal, would have meant that if all of this coal had been carried by the canal, boats with a capacity of 6 681 109 tons must have come empty from Lake Erie.

The writer, in his analysis of the methods by which this transportation of ore and coal could have been effected, believes that the manner he has suggested would have resulted in the least possible empty mileage of the canal-boats.

If the West Virginia coal, amounting to over eight million tons, could have been loaded directly in the canal-boats, at the mines on the Kanawha, Little Kanawha, Guyandotte and Big Sandy Rivers—instead of being loaded on railroad cars, and then dumped from cars into canal barges at the Ohio River—it would have increased not only the empty canal-boat mileage, but would also have increased the total distance each boat would have had to travel in a round trip from Lake Erie, by the distance from the Ohio River to the mines, plus the same distance back to that river.

The increase in the distance each canal-boat would have traveled would have meant an increase in the number of coal barges required, and the fact that the barges would have been loaded for the shallower lock depths in these rivers would have meant a further increase in the number of barges required, beyond what would have been the case if the coal had been loaded in cars at the mines and then dumped from the railroad cars into the canal-boats at the West Virginia towns mentioned. These increases in number and mileage of canal barges for the same total tonnage would have increased the barge cost, over what would have been the case if the barges had been loaded from railroad cars at the

Ohio River for a nine-foot depth; but whether the total cost from the mines to the Lake Erie terminal would have been increased or diminished, would be impossible to predict in advance of actual operation, although the writer believes that when all items that go to make up the total cost, which will be referred to further on, are taken into account, the cost by water would be at least as great as by all rail, and the time would be longer.

In addition to the coal from the various coal districts which was shipped in 1913 to the different lake ports for the upper lake trade, there was a large amount shipped for local use to the Mahoning and Shenango valleys by the railroads serving those districts. The writer has been unable to ascertain the total amount of this coal but has learned that one of the railroads alone carried 3 400 000 tons in that year.

As has been shown, if all the ore coming from the lake ports to the furnace districts, and all the coal going from the coal districts to Lake Erie in 1913, had been transported in canal-boats, it would have required boats of a capacity of nearly seven million tons, going empty from the lake terminus of the canal down the Ohio River for their return load of coal to Lake Erie. It therefore follows that it would have required additional boats with a capacity of 3 400 000 tons, going empty from the two valley districts to the coal districts, to handle the coal carried by one railroad alone, to say nothing of the boats that might have been required to transport the coal carried by the other railroads serving these districts. This would mean that, if the canal had been in operation in 1913 and had transported all the ore and coal in that year, the empty trips would have been the equivalent of over 9165 trips of a single 1100-ton barge—the trips varying in length between the distance from the valley districts to the different coal districts, and the distance from the Lake Erie terminal to Ashland, Kentucky—which does not include the coal carried by the other railroads to the valley districts.

Neither the advocates of the canal, nor any one else, can predict, if the canal is ever built, just how the business of transporting the coal and ore would be carried on. It could be done in any one of three different ways, or by a combination of them all. The owners of the blast-furnaces might have their own

barges for delivering ore to the furnaces, and might contract with coal operators for the return cargoes of coal; or the coal operators might own the boats and contract with the furnace owners for the return cargoes; or the boats might all be owned by one or more transportation companies which would contract with both the furnace owners and mine owners for the transportation of their ore and coal; or all three methods might be in operation at the same time.

If the canal should be built—whether power barges or barges handled by towboats are adopted for transporting the ore and coal—there should always be at the Lake Erie terminal, ready for loading with ore, enough empty canal barges to make sure that the lake steamers which bring the ore from Lake Superior would not have to lie at a pier or wharf waiting to be unloaded. Unless there were, at both the river coal tipples and the piers at the Lake Erie terminal enough extra canal barges to insure there being no delay in loading and unloading either the canal barges or lake steamers, the canal could not possibly be operated with such regularity that every canal-boat, immediately on its arrival at the Lake Erie terminal would have its coal unloaded and be at once reloaded with ore; or that every lake steamer after the ore is unloaded would be immediately reloaded with coal.

Regarding this feature of the traffic the 1907 prospectus of the Canal Company says:

"The greater economy of transfer of ore and coal via the canal over rail is apparent in that the low cost of canal barges to carrying capacity will permit of their being kept in sufficient number at the lake loaded with coal or empty for ore to eliminate the necessity for dockage of either commodity, and the expense incident thereto, but will permit direct transfer at all times."

There can be no question that the interest, operating and maintenance costs of these extra barges would be items that would have to be included in the cost of transportation by the canal. and as this cost cannot be correctly predicted it is another unknown factor that would prevent the actual transportation cost from being known in advance.

With regard to the question of transportation costs by the canal it is pertinent to call attention to the fact that neither in the report of the Canal Board, nor in the paper of the consulting engineer read before the Engineers' Society of Western Pennsylvania, is there anything to show what would be the actual transportation cost by the canal when taking into account the cost of the empty trips. The report of the Canal Board, gives figures of 57.8 cents per ton on ore and 47.8 on coal as the total cost of transportation by the canal to and from the Pittsburgh district. but does not say whether or not these figures include the cost of empty trips. Mr. Stickney, however, in his paper gave the cost per ton-mile for different depths of canal, for freight in both bulk- and package-freight carriers, and in his closing discussion he gave two tables, one for bulk freight and the other for package freight, which tables are headed: "Estimated cost of transportation on the canal based on a season of eight months per year for continuous operations with full cargoes" (PROCEEDINGS, v. 31, pp. 331-332). These tables show how his costs per ton-mile are arrived at. Even if these figures are correct for barges with full cargoes, he does not give any estimate as to how much the costs per ton-mile for *full cargoes* must be increased to cover the cost of the trips with no cargoes.

The writer has gone at some length into the question of how the canal might be operated, in order to show how impossible it is for any one to foretell just how it would actually be operated if it should be built. He does not hesitate to say not only that this uncertainty is one of the causes, if not the main cause, that makes it impossible for any one to accurately estimate or predict unit costs of transportation, but also that the actual costs can be determined only after the canal has been in operation long enough to ascertain *all* the charges that make up the total transportation cost.

WHAT WOULD BE THE TOTAL COST OF TRANSPORTATION BY THE CANAL?

Summarizing, the writer would call particular attention to the items which go to make up the total cost of the operation of the canal:

1. The cost of taking the ore out of the lake steamers and putting it into the canal barges.

2. The cost of taking the ore out of the canal barges and putting it in the furnace stock houses or yards.

3. The cost of taking the coal out of the canal barges and putting it into the lake steamers.

4. The cost of operating the canal barges.

In these four items, operating, interest, insurance and depreciation charges on the loading and unloading machinery, as well as on the canal barges must all be taken into account, no matter whether wharves and their equipment at the Lake Erie terminal are furnished by the United States Government or paid for by the county bond issues.

5. The operating, maintenance and depreciation charges for the canal and locks.

6. The interest and maintenance charges on the Lake Erie terminal harbor which according to the Canal Board will be dredged out by the United States Government. While these harbor charges do not enter directly into the cost to users of the canal, they must be borne by the general public and are actually part of the cost of the operation of the canal—as well as, part at least, of these same charges for the locks and dams on the Ohio River between Pittsburgh and the mouth of the Beaver on the north and Ashland on the south; and also on the Monongahela and Allegheny Rivers; and in addition part of these same charges on the cost of raising to 12 feet the locks and dams already built; and on new dams on the Ohio that are yet to be built as far down as Ashland.

7. Until such time as the total canal revenue equals the operating and maintenance cost of the canal—plus the interest and sinking-fund charges on the bonds to build it issued by the counties—any deficiencies in these two latter charges must also be considered as part of the cost of operation, although the counties through their tax levy must make up the required amount.

The report of the Canal Board says that the Pittsburgh district rate by the canal will be about 57.8 cents per ton on ore and 47.8 cents on coal, as compared with $1.065 on ore and $1.02 on coal by rail. In connection with these figures the fol-

lowing from the final report of the National Waterways Commission, comparing the rates on the Erie Canal with the rates on the railroads, is interesting:

"The average rate per ton mile on the Erie Canal is about 2.45 mills. Comparing this with the average freight rate it is frequently stated that the cost of transportation on this canal is about one-third of that by rail. But no such exact comparisons as these can be made for the reason that the rates compared do not include the same elements of cost. . .

"The average cost of transportation on the Erie Canal includes only the bare cost of conveying the traffic, plus a small return on the equipment used; it contributes nothing for the maintenance, nor does it make any return on the capital expended for the construction and enlargement of the canal. Both these expenses are paid by the State of New York. It has been conservatively estimated that if the canal traffic was charged with a toll sufficient to cover these additional expenses, the rate per ton mile would amount to 8.61 mills. The average ton-mile rate for similar commodities, especially on the railways in New York State is probably not more than five or six mills per ton mile. . .

Any comparison between cost of transportation by rail and by water will be of value only when the two rates include similar elements of cost. Few such comparisons have ever been made."

Does not a consideration of this last paragraph suggest the question, "Has the Canal Board, in the figures given for rates on ore and coal included all the above items, which have been given by the writer as making up the total cost of transportation by the canal?" Without including all these items the estimates of operating costs are not complete, and the average citizen and taxpayer does not realize this fact. Even if all these items have been included in the estimates of cost and operation, the history of all great modern canal construction has shown that these preliminary estimates of cost of both construction and operation, as well as operating revenues, are at best only a guess.

WOULD COAL FROM THE PITTSBURGH DISTRICT BE SHIPPED BY THE CANAL?

The foregoing with regard to cost of transporting coal is on the supposition that in its transportation by canal to the lake ports, there would be no more breakage than by the present method of rail shipment. As regards this question of breakage,

the coal from Logan County, West Virginia, under present methods of combined rail and boat shipment to lake points, has an advantage of about 20 per cent. less breakage than coal from the Pittsburgh district owing to its greater hardness. In connection with the matter of additional breakage of coal shipped by the canal, as well as the amount of coal from the Pittsburgh district that might be shipped by the canal, the following by Mr. S. A. Taylor, past president of the Engineers' Society of Western Pennsylvania, an experienced mining engineer and coal operator, is interesting and also enlightening:

"Regarding the loss of coal due to breakage by handling in what is known as Lake Shipments, I wish to say that about four or five years ago I investigated some shipping problems of this character by way of securing comparative information relative to the breakage of coal from this district as compared with that from the Logan Field in West Virginia.

"I secured the information from two docks located at the head of the Lakes; both of which took coal from our Pittsburgh-Youghiogheny district, and from the Island Creek Coal Company in Logan County, West Virginia. The breakage of LUMP coal loaded at the mine on Railroad cars as ¾ LUMP, under the usual method of screening, shipped via railroad cars to the lakeside, dumped by use of the car unloading device into the vessels, and taken out of the vessels on to the dock at the head of the Lake, and rescreened there into coal cars (this being the usual method of handling coal in Lake Shipments) showed breakage for Pittsburgh coal at one dock in round figures at 32%, and on the other dock practically 34%. The difference in the breakage being due to the difference in the character of machinery used; as the coal was shipped from the same mines.

"The breakage of coal from the Island Creek mines shipped over these same docks was a fraction over 12% in one case, and a fraction under 14% in the other.

"At that time the cost of coal laid down on the docks at the head of the Lakes was $2.78 per net ton. The screenings, or all that portion of the coal passing through a ¾" screen, were sold at an average of $1.75 per ton.

"Taking the average of the breakage of both of these docks it would be close enough for all practical purposes to say that for every three tons of coal shipped from the mines to this district, one ton of this amount at the Northwest was screenings, and was sold at a loss of $1.03 per net ton.

"At these figures the two tons of LUMP coal would have to carry this loss (of 51½c. a ton) over the cost of placing the coal on the docks; before any profit could be secured on the coal shipped in this way, for

use in the Northwest, which is a differential which the district can scarcely stand in competition with West Virginia.

"Regarding the use of the proposed Pittsburgh & Lake Erie Ship Canal for carrying coal from this district for the Lake trade, will say that the amount of coal which is available for transportation by the canal is not very great. There is now very little coal on the Monongahela River, in the 1st, 2nd, 3rd and 4th Pools, close enough to the River to permit of its being loaded directly into barges; and the coal located along the river above the 4th Pool up to the West Virginia state line is nearly all used for local consumption, or in the manufacture of coke. The coal on the river in West Virginia is too soft to permit of the increased handling which would be necessary for Lake use, and would in all probability have to be confined to local steam use along the line of the canal before reaching the reloading stations at the Lake Port.

"The only other coal that I know of that is suitable for Lake Trade is the coal lying back from the river in the 2nd, 3rd and 4th and possibly some in the 5th Pools, of the Monongahela River. In order to make this available to the river and the canal it would necessitate the loading of the coal into railroad cars at the mines for transportation to a river tipple for transfer into barges or boats such as would be able to operate on the canal as proposed.

"This kind of a proposition would require at least two additional handlings of the coal for the Upper Lake Ports, as barges such as would be capable of transportation on the canal would scarcely be safe for transportation by Lake to upper Lake Ports.

"These two additional handlings of the coal would break up the coal much more than the present method of operation would, and if it should not be more than proportional, when this coal was loaded on cars at the head of the lake for distribution into the interior, would make 50% of screenings as against 33% at present.

"Under normal conditions, with the competition which the coal from this district has to meet from the harder coals of West Virginia, would require the lump coal from this district to bear an additional charge 50% more than at present, without the additional charge of hauling the coal from the mine to the river tipple, which would likely be about 25c. per ton for the run-of-mine coal. *All of which I believe would compel the coal companies to continue the use of railroads for lake coal, even should the canal haul the coal at cost, or even for nothing. If this analysis is correct, and I believe it is, the proposed canal so far as its use as a coal carrying agent from this district* [*is concerned*] *is of very little value.*"

In view of this opinion from a mining engineer and coal operator of the long experience of Mr. Taylor, the writer does not think it necessary to go further into the question of whether any Pittsburgh district coal would be shipped by the canal.

With regard to the loss in value of Pittsburgh coal due to breakage in its transportation from the mine when it is loaded in railroad cars and then transferred to lake steamers, the following extract from a book entitled "The Navigable Rhine",* by Edwin J. Clapp of Yale University, is interesting and pertinent to the subject of coal traffic on the proposed Lake Erie and Ohio River Canal:

> "In spite of the cheap transshipment rates of the south German railways, the transshipment territory for coal soon comes to its limit. The best coal will not stand the rough handling it receives at the hands of coal tips and ponderous self-loading buckets. Coal re-sifted on coming out of the barges at Mannheim after its second transshipment—the first being from rail to water at Duisburg-Ruhrort, shows a loss in value of 1.40 marks per ton" [about 33 1/3 cents].

This statement of the actual results of combined rail and water shipments of coal in Germany, agrees almost exactly with Mr. Taylor's statement regarding loss in Pittsburgh district coal shipped to Lake Superior—due to breakage under present conditions of rail and lake transportation—which he says amounts to $1.03 on every three tons shipped, or 34 1/3 cents a ton. This loss would be further increased to about 51 cents a ton, if shipped by way of the canal, due to the breakage resulting from the additional handling in transferring the coal from the canal barges to the lake steamers.

WOULD WEST VIRGINIA AND OHIO COAL BE SHIPPED BY THE CANAL?

The 1917 report of the Canal Board makes no mention of time of trips of barges from the Lake Erie terminal to any other place than Pittsburgh, although the report says the canal will connect the Lake Superior ore fields with the coal fields, not only of Western Pennsylvania, but also of West Virginia and Ohio. The members of the Board must therefore expect that coal would be shipped from the coal districts, and ore to the furnace districts, of these last two states, as this would be the one reason the counties of these states might vote bonds in aid of the con-

*Houghton, Mifflin Co., 1911, pp. 104-105.

struction of the canal. As showing that it is expected that coal from these two states would be shipped by way of the canal, the following extract from the 1907 prospectus should be convincing:

"The Ohio River from a point about 50 miles below the Beaver River penetrates the Bituminous Coal Field for a distance of about 300 miles. The Kanawha and Little Kanawha drain the field of Southern West Virginia, the former being canalized for a distance of 90 miles, and the latter 48 miles. The Muskingum flowing through the coal fields of Southeastern Ohio is canalized for a distance of 84 miles. It therefore, may be said that in the near future, there will be nearly 700 miles of canalized river, *navigable the whole year,* passing through this vast Northern Appalachian field of coal, connecting with the proposed canal and the Great Lakes. Open river navigation obtains at the head of the branches for a portion of each year, which will considerably increase the above mileage."

The writer, however, is of the opinion that little, if any, of the coal from these two states would ever be transported by the canal. This statement may seem hard to believe, in view of the fact that in 1913 there were transported by rail from West Virginia to Lake Erie, for shipment to the upper lakes, 8 275 770 tons of coal; and the Ohio coal district shipped 6 176 624 tons—a total for the two states of 14 452 394 tons.

Now, let us see what would have been the probability of any of this coal being shipped direct from the mines by water, if the canal had been in operation. The canal locks are to be 56 by 400 feet and the depth 12 feet, while the depth of the Ohio River locks is nine feet.

A study of the data given in the following table, of the dimensions of the locks in the Ohio and West Virginia Rivers, referred to in the 1907 prospectus, will give some indication as to whether any of the coal from the mines of these two states, intended for the lake trade, would ever be shipped by way of these rivers and the proposed canal:

TABLE IV

River	Number of Locks	Width Ft.	Length Ft.	Depth Ft.
Muskingum	11	35 8/10	158	4½
Little Kanawha	5	23	125	4
Big Sandy	3	52	158	6½
Kanawha	4	50	270	6
Kanawha	6	55	313	6

The larger locks on the Kanawha are all below Charleston and the smaller ones above.

Is it at all probable, if the canal had been in operation in 1913, that canal barges of 1100 tons capacity, built for a 12-foot canal, would have been sent up any of these four rivers and loaded for their shallow depths; even if the locks were large enough to pass the 1100-ton barges and towboats through them, and if the mines fronted directly on the rivers, so that coal could be dumped from the mine cars into the barges? The writer however, knows that very few of the mines front on these rivers.

The West Virginia coal, over eight million tons, came from south and east of the Ohio River and was transported in cars of the Chespeake & Ohio; Norfolk and Western; Kanawha & Michigan, and Baltimore & Ohio systems.

If the canal had been in operation and this coal were to have been shipped by the Lake Erie and Ohio River Canal route, the writer believes it would have been loaded at the mine in railroad cars, as is now done, and that enough coal from the cars of the first two roads would have been transferred to the canal-boats on the Ohio some place near Huntington or Kenova to load them to full capacity; and then sent up the Ohio River to the mouth of the Beaver River, which is the southern terminus of the canal—a distance of over 287 miles—then 101 miles by the canal; or a total water haul of about 389 miles. The coal from the cars of the Kanawha & Michigan would probably have been transhipped to canal barges at Point Pleasant, with a river haul of 236 miles to the southern canal terminus, making a total water haul to the lake terminal of about 337 miles.

The coal for the lake trade from the northern West Virginia districts in which Grafton and Clarksburg are situated—which in 1913 amounted to 2 248 546 tons—is now all loaded in railroad cars at the mines and goes mainly by the Baltimore & Ohio R. R., by way of Parkersburg, New Martinsville, Moundsville or Wheeling, to Benwood and thence to Lorain, Sandusky and Fairport, where it is transferred to the lake steamers. The writer also believes that, if this coal was to have gone to Lake Erie by way of the proposed canal, it would also have been loaded in cars at the mines and would have been transferred to make full loads

for the canal barges at some of the West Virginia towns last mentioned.

The writer, however, does not believe that, if the canal is ever completed and in operation, any of the West Virginia coal would be transferred to the canal-boats—even after all the locks and dams on the Ohio between Ashland and Beaver are completed—on account of the uncertainty of a sufficient stage of water at all times during the shipping season, and also on account of interruption of traffic when the river is at flood stage. He further believes the long additional water haul, the extra handling of the coal, and its consequent breakage—even of the Logan County coal, although its breakage is less than that of the Pittsburgh district coal—would prevent any of the West Virginia coal from being carried by the canal. It is also improbable that any of the coal from the mines in the eastern Ohio coal fields, located in Stark, Wayne and Summit Counties on the north, which is now all loaded in railroad cars, would be carried south to the Ohio River and then by canal barges to the Lake Erie terminal—this for the same reasons as those given regarding the West Virginia coal, and also because when once loaded on the cars it can go by rail to lake ports in much less time and with less breakage than by the Ohio River and canal route. The writer also believes that coal which could be loaded directly from mine cars into the canal-boats on the canalized Muskingum River, would still go to Lake Erie by rail instead of by water—even if the boats could be loaded to their full capacity instead of for a depth of only 4½ feet—as the comparatively small number of mines that might use the water route are located in Muskingum County, south of Zanesville. The distance by rail from these mines to Sandusky averages about one hundred and forty miles, while by the all-water route to the Lake Erie terminal of the proposed canal it is about three hundred and twenty-five miles, and the boats would have to pass through 36 locks—10 in the Ohio River and the remainder in the canal.

There is also a large number of the Ohio mines between Zanesville and Athens, in Muskingum, Perry, Hocking, and Athens Counties. Corning, which is 197 miles by rail from Toledo, is about the center of this district. If the coal from

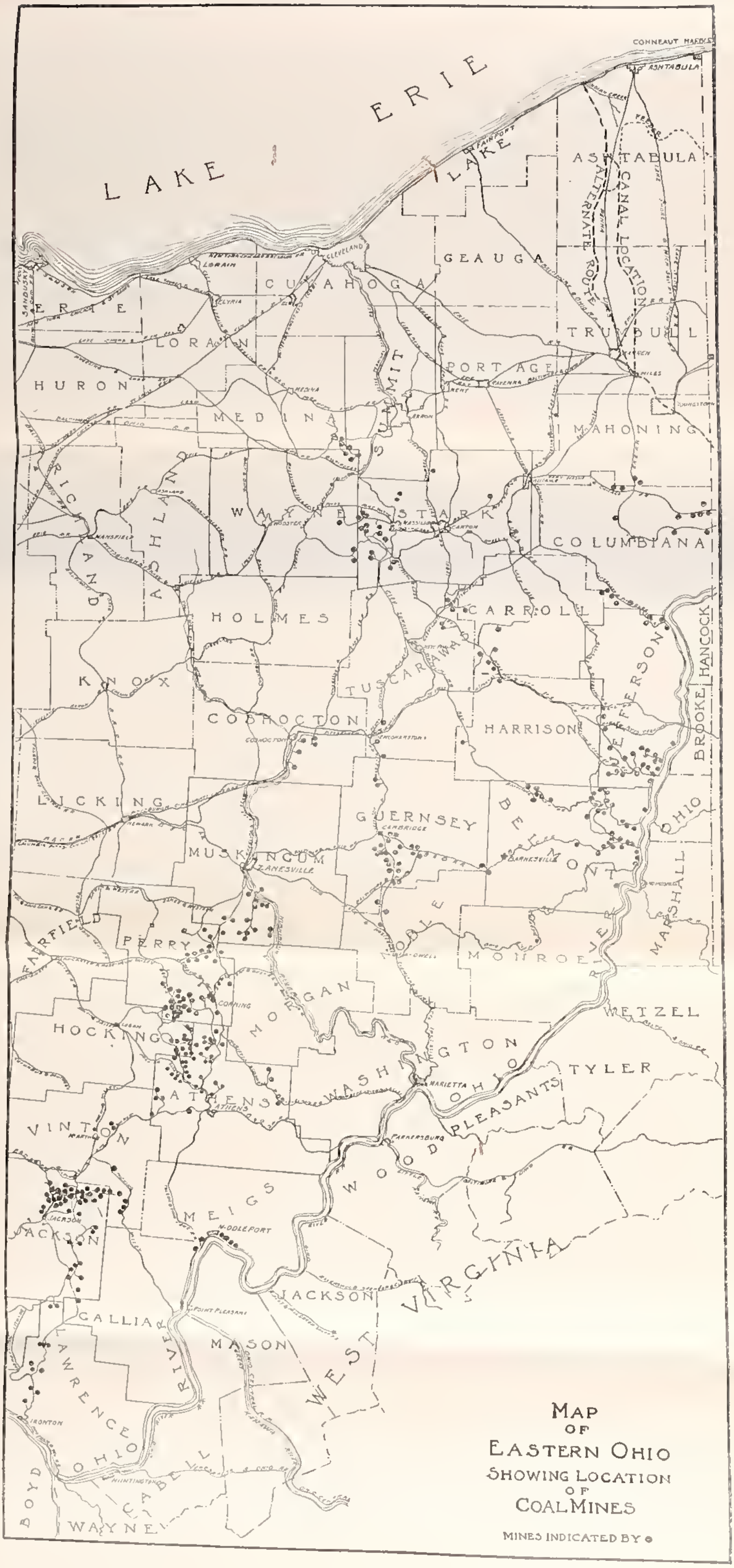
these mines was to be shipped by way of the proposed canal it would have to go by rail to Middleport on the Ohio River, a distance of about eighty miles, where it would have to be transferred to the canal barges; then up the Ohio through 18 locks, a distance of 323 miles, to Beaver; then by the canal through 26 locks, a distance of 101 miles, to the lake terminal—a total of 44 locks, a total water haul of 424 miles, and a total distance by rail and water of 504 miles; or over two and one-half times the average direct rail haul from the mines to Toledo. The time by water from Middleport to the Lake Erie terminal at a speed of 62 miles a day, would be almost seven days, and the barges would have to pass through 44 locks.

In addition to the Ohio districts already mentioned there is in the eastern part of Jackson County a large number of mines, the output of which goes largely to Lake Erie for the upper lake trade. If this coal was to go by way of the proposed canal it would have to be loaded in railroad cars at the mines, and go by rail to Ironton, an average distance of about sixty miles, where it would be transferred from the cars to the canal barges with the resulting increase in slack. It would then go by way of the Ohio River and canal through 50 locks for a distance of about five hundred and sixty miles to the Lake Erie terminal.

This coal is now shipped by rail and travels an average distance of about two hundred miles from the mines to Lake Erie where it is transferred to the lake steamers in less time and with one less handling, and consequently less breakage and greater value, than would be the case if shipped by the Ohio River and canal route.

Is it at all probable—taking into account the longer distance, lower speed and consequently longer time, as well as the extra handling and breakage of the coal caused by the transfer from the railroad cars to the canal barges and the interruption to traffic sometimes caused by high and sometimes by low water—that any of this Ohio coal would be shipped by the canal route?

These are some of the reasons why the writer believes the mines in the counties of the West Virginia coal districts, served by the roads mentioned, as well as those in the counties of the


LAKE ERIE
ASHTABULA
CANAL LOCATION
ALTERNATE ROUTE
LAKE
GEAUGA
CUYAHOGA
ERIE
LORAIN
HURON
TRUMBULL
PORTAGE
SUMMIT
MEDINA
MAHONING
WAYNE
STARK
COLUMBIANA
RICHLAND
ASHLAND
HOLMES
CARROLL
KNOX
TUSCARAWAS
JEFFERSON
HANCOCK
BROOKE
COSHOCTON
HARRISON
LICKING
GUERNSEY
BELMONT
OHIO
MUSKINGUM
MARSHALL
FAIRFIELD
PERRY
NOBLE
MONROE
MORGAN
WETZEL
HOCKING
WASHINGTON
TYLER
ATHENS
PLEASANTS
VINTON
WOOD
MEIGS
JACKSON
WEST VIRGINIA
GALLIA
MASON
LAWRENCE
OHIO RIVER
BOYD
CABELL
WAYNE
CLEVELAND
SANDUSKY
ELYRIA
MANSFIELD
CANTON
MASSILLON
WOOSTER
AKRON
RAVENNA
WARREN
NILES
ZANESVILLE
CAMBRIDGE
BARNESVILLE
CORNING
MARIETTA
PARKERSBURG
ATHENS
JACKSON
POMEROY
POINT PLEASANT
IRONTON
HUNTINGTON
ASHTABULA
CONNEAUT
FAIRPORT
MAP OF EASTERN OHIO SHOWING LOCATION OF COAL MINES
MINES INDICATED BY •

eastern Ohio coal fields, would not use the canal if it should be built.

There are, however, two counties (Belmont and Jefferson) in Ohio, and three (Hancock, Brooke, and Ohio) in West Virginia, which are situated not far south of the southern canal terminus at the mouth of the Beaver River, from which the mines that front on the Ohio River might be able to ship coal by the canal. The writer believes, however, that even these counties would not ship coal by the canal route on account of the extra breakage caused by the additional handling—even if the increased length of haul and the longer time required to reach the lake terminal, over that required by rail, did not prevent shipment by the canal.

An inspection of the map (Plate 2) showing the coal mines in Eastern Ohio, reproduced from one of the reports of the Geological Survey of Ohio, the writer believes should be convincing as to the correctness of his reasons for believing that the prophecies by the canal advocates of the large amount of Ohio coal that would be shipped by way of the proposed canal will not be fulfilled if the canal should be built.

These reasons lead to only one conclusion: that little, if any, Ohio and West Virginia coal would be shipped by the canal; and if Mr. Taylor's opinion—that the extra breakage of Pittsburgh district coal "would compel the coal companies to continue the use of railroads for lake coal, even should the canal haul the coal *at cost, or even for nothing*"—is correct, then the canal would obtain practically no coal business.

If the canal would obtain no coal traffic, is it at all probable that it would obtain any ore traffic, when it would mean that the canal-boats that carry the ore south would have to return to the Lake Erie terminal empty; or that the furnace plants would go to the expense of changing the arrangement of their plants, and installing the machinery for transferring the ore from the boats to the stock houses, relying on the canal to bring their yearly ore supply from Lake Erie in eight months, and in some years probably in only seven months?

With regard to the general question of water transportation, Mr. Frear, member of the Rivers and Harbors Committee, United

States Congress, has well summed it up when he says, "Without railway ownership and control by the Government, it is doubtful if one great inland waterway success would be recorded." This statement of Mr. Frear is, in the opinion of the writer, as true regarding water transportation in the United States as it is in England, Belgium, France or Germany. He also believes it will prove true of the Lake Erie and Ohio River Canal, if it should be built. In fact, the members of the Canal Board actually admit it, indirectly, when they depend on the United States Government to build the Lake Erie terminal harbor, and not only to maintain it, but also to build and maintain the locks and dams in the terminal harbor at the southern end—for that is what the three rivers practically are—on which the coal for the lake trade is to be transported before entering the canal, and on which are located most of the blast-furnace plants where ore would be unloaded.

CONCLUSIONS

The writer believes:

1. That he has discussed the question of whether the Lake Erie and Ohio River Canal should be built, in accord with the suggestion of the president of the Canal Commission "that it should not be used as a political football, but considered as an economic question only."

2. That as the president of the Canal Board has attached so much importance to the fact that ten great waterway engineers have pronounced the canal feasible, he is warranted in calling attention to the following facts:

Twelve members of the National Waterways Commission, seven of whom were United States Senators and five of whom were representatives in Congress, have signed their final report in which they have expressed the opinion that it might be more economical for the mines in the Monongahela basin to patronize the railroads that lead direct to the mine, than to use the canal; and that *the amount of coal which the canal might carry has been overestimated.*

In addition to the opinion of the members of the Commission themselves, there is expressed by Colonel Newcomer, one of the

ten waterway engineers referred to above, in his report to the Commission, *his own doubts as to whether there would be traffic enough to provide a revenue commensurate with the expense of the work.*

There is also the opinion of Mr. S. A. Taylor, E. M., consulting engineer and coal operator, that the coal companies would continue to use the railroads *"even should the canal haul the coal at cost, or even for nothing."*

Prof. H. G. Moulton of the Department of Political Economy of the University of Chicago said in a letter to the writer:

> "I have always felt that the Lake Erie & Ohio River Canal had a better chance of success than any other project in the country, but that the odds were overwhelmingly against it. I have . . . no hesitation in saying that *you raise an almost unanswerable presumption against the project."*

DISCUSSION

Mr. Lee C. Moore:* The project under discussion presents an opportunity for statements plainly indicating that the entire project, though visionary, has a subtle purpose. Canals can be used as waterways, north of Mason and Dixon's Line, only about eight months in the year; but as a method of disbursing funds for various purposes, they can be utilized the entire year.

It is a fact that canals or waterways—if provided by nature, or if the topography is such that they dig and possibly maintain themselves—are a commercial success. This is frequently the case abroad, notably in Holland, where they are also utilized as line fences between farms, ditches for irrigation, and other purposes.

There was an acknowledged necessity for several ship canals on this hemisphere—the Welland, Sault Ste. Marie, Panama and possibly others. The Erie Canal was a necessity at the time it was projected because of no facilities for building railroads. At present it is a splendid waterway for the disbursement of public funds for various purposes, such as fixing up fences. It is understood that the state of New York is now disbursing the second hundred million dollars within the last 18 years; and some of this can be used for the purpose just mentioned.

The writer believes the above statements can be substantiated, and if the Lake Erie and Ohio River Canal is built it can be used for the purpose indicated in the preceding paragraph.

Mr. George M. Lehman:† The writer, having been closely connected with the Lake Erie and Ohio River Canal project, has read with peculiar interest the paper on that project by Mr. Wilkins.

The author practically places the engineering profession, so far as this canal is concerned, in the position of having met utter defeat; considering it impossible to reach even the customary preliminary conclusions, not only as to estimates of cost of construction and operation, including methods and cost of transportation, but as to economic and financial features—arguing, in fact, that nothing of any importance in this particular work can be even

*Lee C. Moore & Co., Inc., Engineers, Pittsburgh.

†Chief Engineer, Lake Erie and Ohio River Canal Board, Pittsburgh.

approximately determined. His enunciations are based on several brief reports, together perhaps with certain personal knowledge of conditions and the opinions of several men—not engineers nor of this locality—who have never given the matter comprehensive and unbiased study.

If it is impossible to arrive at satisfactory preliminary plans and costs for this canal, then it cannot be done for other large projects, often with many involved problems—for instance, water powers, flood control, railroads, subways, etc.

The canal project consists of river canalization; the cutting of a ditch; building of locks; modifying and changing certain bridges; water-supply. All these things are accomplished all over the world, daily, and with this canal it is only a matter of having the various parts properly planned and estimated, and brought into harmonious relationship. The project is, of course, a large one with many important problems, but all surmountable, and the writer does not for one moment subscribe to the idea that it is beyond satisfactory accomplishment of the engineer, or others qualified to judge.

Mr. Isham Randolph, of Chicago, who acted as one of the consulting engineers—a man of more than national reputation, having been a consulting engineer at Panama, builder of the Chicago Drainage Canal, and in his time connected with many affairs of great prominence, including railroads—said: "I know of no great public improvement that has received the careful, comprehensive study that has been given to the proposed Lake Erie and Ohio River Canal." This includes the economic outlook.

Mr. Frederick P. Stearns, of Boston, another of the consulting engineers, is a man of national reputation, especially in hydraulics. He also acted on the Panama Canal plan, and has been connected with many water-supply and other related works throughout the country.

Mr. H. McL. Harding, of New York, well known in a number of cities as an expert on terminals and docks, advised on the terminal question for the canal.

The economic study was made by Dr. J. T. Holdsworth, of the University of Pittsburgh, and was most exhaustive.

It is needless to say that these men are particularly well qualified to act in a consulting capacity for this canal.

It is usually considered that no engineer can give just or adequate judgment on an engineering project, unless he is especially qualified and has been engaged for some time directly upon the work. Reliable opinions or conclusions certainly cannot be based upon generalities. In this opinion the writer feels quite sure the author would concur, after proper deliberation, as he is a man of much engineering experience.

COST, AND CERTAIN PHYSICAL FEATURES

It is gratifying, however, that the author has elected to pronounce the canal feasible physically, and it will henceforth be so considered; but it is really aggravating, after it is decided that the canal is physically feasible, that the engineers are not permitted even to think that they can do anything worth while with it, even in the way of trustworthy reports.

No one has supposed, of course, that the cost of the canal has been determined to the exact dollar, but only that it has been accomplished as close as is usual with engineering work receiving careful and exhaustive attention. The surveys and field investigations have been very extensive, and in the recent office work alone the reduction of the field notes, including topographic and hydrographic features, covers 300 large sheets, besides numerous other maps and many profiles. The books containing the estimates in systematic form, make a pile over four feet in height.

The Board of 1914, desiring to have outside opinion in the new estimates for the canal, and to have previous work properly investigated, secured the services of Mr. G. F. Stickney, who had been engineer in charge of design and construction of a division of the New York State Barge Canal. The writer has absolute confidence in the excellent work done by Mr. Stickney in estimating the cost of this canal; the work being based upon recent and earlier surveys and plans of unusual thoroughness.

In round figures, as given in the report of the board of last year, the construction cost in accordance with the present plan, including water-supply and all other costs, for the canal of 140-foot minimum bottom width, 12-foot depth, is $65 000 000. With things running in a fairly smooth manner, the whole work should

be completed in five years. As to the full cost, including interest during construction, a close approximation based on good judgment can be made. It will not be as high as thought by the author.

With regard to the bridges—about which the author feels rather perturbed, seeming to fear that some had been dropped out —it may be said that those requiring attention have been properly accounted for in accordance with the estimate.

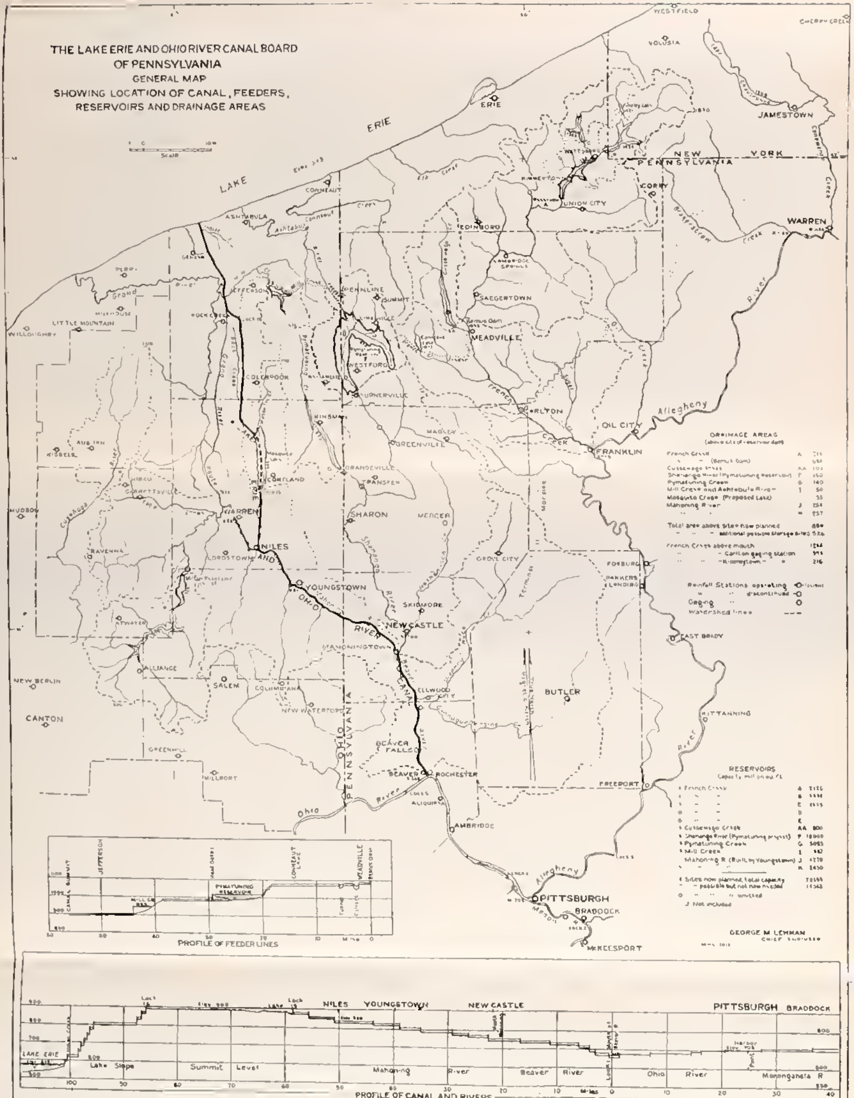
Certain small modifications were made in the estimate, brought about by using the large lock, 56 by 400 feet, in connection with a 140-foot channel finally adopted by the Board. It was thought that the channel could be widened when business demanded. The plan of water-supply was modified by Mr. Stearns and the writer. This concerned consideration of the upper French Creek, Cussewago, and Mill Creek reservoir sites. French Creek site is particularly valuable for conserving high run-off now going to waste and frequently causing damage. This is one of the sites found by the writer during the investigation for flood control, in 1909-1910. The construction of this site would be of material benefit in the way of flood relief to Meadville and to other local points, and would also contribute towards relief at Pittsburgh.

The paper expresses fear that the farmers of the French Creek valley will suffer for need of water, during the season of low flow, due to the storage reservoir. It therefore will be explained that, naturally, in the operation of the plan, the low flow which may obtain at times during the summer will be materially augmented by a well-regulated, serviceable discharge, and it is natural to suppose that the farmers, as well as others, will be much benefited. Any attempt to use low flow to the discomfort of others would be injudicious and is not for one moment considered. Care is taken to improve present conditions. Under the circumstances "pay," as mentioned in the paper, has not been included. Relative to the right to take water from French Creek, the acts of the various states provide for this, but of course it must be done in proper manner. Attention is called to the fact that a national charter, of some years ago, also granted the right. Moreover, lawyers, highly fitted for opinion, did not find anything of a detrimental nature.

TRAFFIC

It is not possible to ascertain, in advance, the exact character or the amount of the tonnage that would be transported, either as to through movement or between intermediate points. As with many other large business propositions, however, estimates may be made with fairly accurate results. These estimates are based on the demand for additional transportation facilities, and the present business and future possibilities of the canal zone and the great tributary surrounding country, in all of which tonnage is increasing by leaps and bounds, the greater part of it being particularly well suited to carriage by water. In such a region, forming the very heart of our country, it is obvious that where only one form of transportation now exists, any new method which is of an economic nature will be eagerly sought. It must also be granted that undoubtedly the presence of the canal improvement would have an enormous creative force and influence on production and movement, not only in the canal zone, but throughout a large part of the country, from New York to the Great Lakes and on down to the Gulf of Mexico, including the connecting water courses. This inland link is so situated that it will combine commercial with military value, providing an east and west as well as a north and south movement.

The writer has no visionary ideas about this project, at least to any material extent—always having tried not to imbibe them—and he fully realizes that, on account of physical conditions and inadequate business, great and serviceable canals for commercial purposes cannot be built everywhere in the world with satisfactory return on the investment. Here, however, where nature favors with natural resources and advantageous physical conditions, together with tremendous and constantly growing forces built up by man, we have a proposition to connect—by a short line of travel, 101.5 miles long, only 50 miles of which consist of actual ditch—two of the most important bodies of water in the world. If water transportation would be a failure here, with the potentialities of great industrial mobilization, there is no use of seriously thinking of it in any other part of the world. There are advantages and disadvantages, but the former by far overtop the latter. It has just come to mind that the Manchester Canal has



THE LAKE ERIE AND OHIO RIVER CANAL BOARD
OF PENNSYLVANIA
GENERAL MAP
SHOWING LOCATION OF CANAL, FEEDERS,
RESERVOIRS AND DRAINAGE AREAS
LAKE ERIE
ERIE
JAMESTOWN
NEW YORK
PENNSYLVANIA
CORRY
UNION CITY
WARREN
EDINBORO
SAEGERTOWN
MEADVILLE
PENNLINE
SUMMIT
WESTFORD
ASHTABULA
CONNEAUT
LITTLE MOUNTAIN
WILLOUGHBY
COLEBROOK
KINSMAN
GREENVILLE
FRANKLIN
OIL CITY
Allegheny
WARREN
NILES
YOUNGSTOWN
SHARON
MERCER
GROVE CITY
NEW CASTLE
MAHONINGTOWN
ALLIANCE
SALEM
CANTON
NEW BERLIN
RAVENNA
BUTLER
KITTANNING
FREEPORT
EAST BRADY
BEAVER
ROCHESTER
AMBRIDGE
PITTSBURGH
BRADDOCK
McKEESPORT
OHIO RIVER AND ERIE CANAL
OHIO
PENNSYLVANIA
DRAINAGE AREAS
RESERVOIRS
GEORGE M. LEHMAN
PROFILE OF FEEDER LINES
PROFILE OF CANAL AND RIVERS
Lake Slope
Summit Level
Mahoning River
Beaver River
Ohio River
Monongahela R.

been referred to, and attention should be called to the fact that there is but little comparison between the two projects and the immediate countries in which they are located.

As already intimated, the tonnage possibilities will be influenced by character, accessibility, cost, and handling in transportation; also to some extent by the time of movement, although with bulk or low-class freight the latter is not of great importance.

To give some idea as to the great amount of tonnage moving several years ago on the Great Lakes, and by rail and river in the canal zone, the following figures are given, in round numbers:

Great Lakes	100 000 000
Pittsburgh district	173 000 000
Wheeling	25 000 000
Youngstown	26 000 000
New Castle	8 400 000

Dr. J. T. Holdsworth, in his economic study of 1915, considered that over 80 000 000 tons of miscellaneous freight, consisting of iron ore, coal, iron and steel products, limestone, sand, gravel, cement, brick, lumber, etc., were particularly available for the canal.

From data just received, the writer has been able to ascertain and tabulate the number of furnaces, and the ore transport possibilities, by basing figures on the pig-iron capacity of the furnaces. Only the furnaces located *on* the rivers have been considered. Shipment figures indicate that the method employed appears very conservative. It is understood that the furnaces frequently run at capacity, at least for a portion of the year.

District	No.	Ore consumption at capacity. Tons	Amount estimated for water haul. Tons
Pittsburgh, above mouth of Beaver..	57	18 126 000	
Youngstown & New Castle................	26	7 821 000	
Wheeling (including Midland)........	15	3 922 000	
Ashland & Ironton...............................	9	1 131 000	
Total..	107	31 000 000	16 100 000

The above is estimated on the basis that about 80 per cent. would be shipped in normal times. Also that about 50 per cent. of the United States Steel Corporation ore, 85 per cent. of the ore of independent concerns of the Pittsburgh district, and 75 per cent. of the ore of the other localities would pass through the canal. These figures are undoubtedly very conservative.

Scrutiny of about three dozen furnace sites, taken at random, shows that the stock sheds or piles are located an average distance of about 312 feet from the river edge, and that no difficulty would be experienced in planning machinery for conveying from the dock. Coal-conveying machinery is now in operation at a number of places—with more contemplated.

As to coal, contrary to the opinion of the author, there appears to be still unmined, close to the Monongahela alone, a very large quantity stretching along the river from a point some distance above McKeesport, Pa., into West Virginia. Unquestionably, the amount is sufficient to provide considerable tonnage for water haul for a number of decades. As is well known, great quantites of coal constantly go by water to manufacturing plants; and a material increase will take place, as some of the large users are now being equipped to handle by river. One large local concern covers a distance of over 100 miles, one way, from the tipple on the Monongahela to the mill on the Ohio. Another interest has for many years used the river with great regularity, transporting several millions of tons annually. A company in the Youngstown district recently bought a very large coal area fronting on the Monongahela and, it is understood, will eventually ship close to 2 000 000 tons annually. The supply will last many years.

Judging from the remarks of users, there would undoubtedly develop a very large water-borne coal traffic to works on the canalized portion of the canal, alone. The number of handlings would be the same as at present on the local rivers. The writer will not undertake to say just how much would move entirely through the canal, to lake points and to the New York State Barge Canal, but the aggregate, both through and local, might easily exceed 15 000 000 tons, annually, several years after opening the canal.

The annual cargo coal shipped to the Great Lakes from the Pittsburgh district and the upper Ohio River amounts to about 27 000 000 tons, nearly three-quarters of which is mined in Pennsylvania and West Virginia.

Tax returns on coal areas in townships fronting on both sides of the Monongahela River, only to the West Virginia line, show that there were over 1 400 000 000 tons in 1915. The estimate of a large operator for both sides of the Monongahela and for a distance of five miles back, extending from a point above McKeesport, Pa., to the West Virginia line considerably exceeds the above amount. On the lower Allegheny River many millions of tons remain unmined. This is also the case along the Monongahela in West Virginia.

In 1915, five important counties of southwestern Pennsylvania produced over 90 000 000 tons. Assuming that the unmined coal in the country closely bordering the Monongahela, only to the West Virginia line, does not exceed 1 400 000 000 tons, and that 15 000 000 tons will be mined annually from these areas, all going by water into or through the canal, the movement would continue for about 95 years. The coal of West Virginia and southern Ohio, available for water haul and accessible to the upper third of the Ohio River, including its navigable tributaries, must not be overlooked.

It should be mentioned that the controlling depth over lock sills, for the lower 57 miles of the Monongahela, is close to 8.5 feet, but on the upper portion, to the head of the stream, it is about 5.5 feet. Permits for dumping in the channel now specify that a depth of not less than 12 feet must be maintained. On the Ohio, in a distance of about 320 miles, nearly 70 per cent. of the lock sites have been completed. Over 80 per cent. of the number have a minimum depth of 11 feet over the sill; one has 12 feet. If occasion demands, there will evidently be no difficulty in having the Government increase the depth to 12 feet on the Monongahela and Ohio. Low water troubles are not expected, to any material extent at least, with the Ohio River pools completed. A considerable discharge will be sent down by canal operation.

The author speaks of one very important matter, and that is the degradation caused by handling. There is, of course, an

economic limit. When the number of handlings exceeds what is now usual it is hard to say what will take place. Several ideas occur, however—one of them with regard to the method of transportation by water, and this may largely or completely offset the trouble. That little or no coal will go by water does not seem reasonable.

MODES AND COSTS OF TRANSPORTATION

The writer must differ with the opinion that it is impossible to make a fairly close determination of the method and cost of transportation. Based upon experience on existing waterways, together with special and painstaking study applied to all known conditions of the proposed canal, there is no doubt as to the accomplishment of results reliable enough for the present purpose.

It must be admitted beyond controversy that as a general proposition water transportation, on adequate waterways, is cheaper than any other form; furthermore, that it is well suited to care for certain commodities in large bulk. There is little difference between slack-watered rivers and artificial canals when the latter are of ample size and physically favored. Local experience shows that river transportation is of great economic value.

Canals, when fortunately located and well designed, even though partially formed by natural streams, may have facilities for almost complete water control, and may therefore experience less disturbance and fewer troubles from currents and weather conditions than is the case with rivers. The Lake Erie and Ohio River Canal will be much like this, and in some respects maintenance will be nearly as economical as for several of our local rivers.

This project is not to be regarded in the light of a contest with the railroads; such an idea is unnecessary and unwise. Looking at it in the broad sense, it will not be injurious; and with operation under a well-organized system of co-operation and co-ordination it will be highly beneficial to all interests. It is understood that railroads operating along waterways, and touching harbors and adequate terminals for interchange of traffic, invariably pay better than those remote from such advantages.

In general, railroads have a wider area of distribution; but canals like the one in question, where great density of industrial establishments obtains immediately along the route, can also operate with broad and effective results under modern businesslike management and proper co-ordination. Modern business methods and efficient equipment are, of course, essential for this canal, and nothing else is to be considered.

Concerning the type of craft for handling the traffic, no attempt has been made to form any exact plan of boat, or of movement, at this time. Complete standardization, if finally desired, can come only during the process of early development after the opening of the waterway. What was done was to make an investigation of types now in use or proposed, with such modification as appeared advisable and best suited to the canal. Costs were worked out accordingly.

Reference is made, by the author, to the paper of Mr. Stickney, read before this Society in 1915,* in which costs were given for a self-propelled barge and gas-producer; and as practically nothing was done relative to the towed-barge fleet system, and as this is a most effective and economic method, the writer considered that it must have proper attention. Hence, the fore part of last year he made an investigation which led into developing costs embracing both towed fleets and self-propelled barges of several types. The commodities considered in this work did not extend beyond such articles, for instance, as iron ore and coal.

For another reason it was advisable to consider the fleet method—on account of the large existing, and constantly increasing, investment in steel barges used on our local rivers directly connecting with the proposed canal. The horizontal dimensions of the canal locks are to be the same as those of the Monongahela River, which are 56 by 400 feet. Such a lock would pass at one lockage a towboat and three steel barges, capable of carrying 3600 tons at capacity. The draft is limited to 10 feet. While the lock has this capacity, the channel of 140 feet in width and 12 feet in depth, finally chosen by the Board, necessitated a slight reduction in the size of boat, and limited its capacity to 1100 tons instead of 1200.

*Proceedings, v. 31, pp. 285-333.

It is necessary for effective results that the water section of the channel be made not less than four times the submerged area of the boat or fleet formation, in cross-section.

The American Bridge Company, through the kindness of Mr. L. J. Affelder, provided the cost figures of all the barges used in the computation.

For the purpose of arriving at the average cost, a trip of 138 miles was taken, from Braddock to the entrance of the canal at Indian Creek, Lake Erie. Space and time cannot now be taken to give details as to the many features entering into the study; they are entirely too numerous and complicated, but many interesting things were disclosed. The unit barge fleet, as anticipated, was found cheaper in cost of movement than the self-propelled barge; partly for the reason that the towboat of the former—in a trip to Youngstown, for instance—is inactive only about 5 per cent. of the time; while for the latter the period of inactivity amounts to about 35 per cent., due to the time of loading and unloading.

The self-propelled barge of 340 to 350 feet in length, considered by the marine engineer and used by Mr. Stickney, was considered by the writer as being far too unwieldy, even though large turning basins were specially built for them. This barge, therefore, was considerably reduced in size, but the cost figure was found considerably greater than that used for the one of greater length—even for normal-time prices. This, together with additional items for fixed charges and operating expenses, brought the movement cost to a higher figure than that given by Mr. Stickney. In the recent work the horsepower of the boats was considerably increased—in one case 100 per cent.—and allowance made for a large additional number over that in regular service. The speed that had been used in the computations of the above large barge was found too great. This was checked at the naval plant in Washington. The speeds used in the latest work were from 5 to 6 miles per hour for the rivers between locks, and 4¼ miles for the canal, including the Mahoning River.

The study included movements and types of boats as follows:

1. Barge fleet of seven barges—total capacity 8400 tons—towed by a large tunnel propeller, or stern-wheel towboat, from

Braddock to the mouth of the Beaver River. Small fleet of three barges—capacity of 3600 tons—towed by a smaller towboat, tunnel propeller, from the mouth of the Beaver River to Indian Creek.

2. Small fleet, same as above, from Braddock to Indian Creek.

3. Self-propelled barge, gas-producer, from Braddock to Indian Creek.

4. Self-propelled barge, semi-Diesel, from Braddock to Indian Creek.

In this work computations were made; first for a channel of 180 feet in width, 12 feet in depth; next for 140.5 feet and 13 feet, and lastly for 140 feet and 12 feet.

It was found that the through movement cost for no. 1 was only slightly lower than for no. 2, and that in the small channel the cost of no. 2 was but little increased over that when working in the large channel. As stated, the fleet barge capacity for the small channel was made 1100 tons. The operation of the semi-Diesel proved to be more expensive than operation by the gas-producer. The cost per trip for the small fleet in the 140-foot channel amounts to 13.8 cents, including 20 per cent. profit. To make allowance for more or less empty trips, computations were based on 85 per cent. capacity. It might have been more conservative had it been 80 per cent. but the increase in cost would hardly be appreciable.

As given in the report, the rate for coal is 47.8 cents, including movement, toll, one charge for transfer, and one for degradation, at the lake end. For iron ore, it is 57.8 cents, including movement, toll and two transfer charges.

Maintenance of the harbor at Lake Erie or of the local rivers was, naturally, not charged in, as such a thing would not be in keeping with what is done, at least throughout this country, in similar cases. How would a direct charge be made for this on an equitable basis? One might just as well expect to have included in an ocean-vessel rate a charge for governmental expense of maintaining a harbor channel; or to have inserted in a rail rate an item for maintaining a harbor with which there is a connection and from which benefit is derived.

In the analysis, very careful consideration was given to all factors entering into the problem; such as character of river and canal channel, speed in pools, and time at locks, including maneuvering in approaches, in channel, passings, making up tows, and detention at terminals. In the transfer charge there was also included an expense for extra handling machinery during times of periodical heavy business. No attempt was made to determine just what sort of organization will conduct the transfer business or what its connections will be.

In addition to all ordinary charges for overhead and operation, there were included such items as insurance on life and property, taxes, management, etc. The estimated cost of boat operation is very much the same as that computed for the company in 1906, but it is a matter of personal knowledge that the recent work was more thoroughly done than that of earlier date.

To be conservative, it was considered that operation, due to weather and other conditions, would continue for only 28 days of the month. This suggests mentioning that the lake season is nearly eight months, but it is quite problable that to the Youngstown district the canal would be operated for a period of not less than 10 months. The author has found comment on the likelihood of long detentions, due to breakdowns, with a single-lock system. In this connection, based on the experience of those who for some years have had the care of locks on the Monongahela River, it is the opinion that not exceeding five to six days, on the average, would be lost in a canal season. It must not be overlooked that there will be from two to four months, each year, in which to make repairs and put things in order; and this would also apply to the entire canal plant. Important parts of the structures would always be on hand for immediate placing. In the single-lock system provision would be made for adding another lock, alongside, when business demanded.

Finally, it was found that the small fleet would make 50 round trips per season, and with this system, working at 85 per cent. capacity, with the canal at capacity, 25 290 000 tons could be carried; with the self-propelled gas-producer barge, 47 round trips and 21 530 000 tons for the season. As electric lights would be installed, the canal could work 24 hours of the day, and in this

time it was estimated that 41 lockages, under normal operation, would be made. The number of actual working days per season would be about 224.

Concerning the time of movement by water and by rail, the prevailing impression is that the former is slower than the latter, but when considering the ordinary freight train with its numerous yard and side-track detentions, this is mostly not the case. The writer has made a comparison of various cases of low-class, rail and water haul, with findings in favor of the latter. In the absence of congestion or other troubles, rail is quicker with high-class freight train service on ample roads.

In the paper under discussion there seems to be nothing which would indicate a realization of such a thing as a transportation problem, which is now facing practically the whole country, particularly in highly developed industrial sections—this notwithstanding the fact that the railroads, under the most able management, are doing their utmost to cope with conditions.

The remarks of others may be of interest, and some are given in a fragmentary manner:

Colonel H. C. Newcomer, Corps of Engineers, U. S. Army, in his report of 1911, said:

"Under the existing freight rate for the transportation by rail of iron ore and coal between the Pittsburgh district and Lake Erie, it is believed that there will be sufficient traffic on the canal in these two commodities to pay a reasonable rate on the capital required for the construction of a canal of suitable dimensions.

"It is thought that the estimates of tonnage . . . 21,000,000 tons . . . and about 13,000,000 tons of coal . . . are conservative estimates for the present movements of these commodities.

"There will, of course, be indirect benefits attending the construction of the canal that might go far toward paying its cost of construction."

(This immediately follows a paragraph quoted by the author.)

Lieutenant Colonel Henry Jervey, Corps of Engineers, in a recent address, said:

"That internal waterways constitute a valuable military asset, seems to be an obvious fact. It is difficult by extended argument to make more emphatic the truth of the statement. . . And the development of waterways, in which a degree of inspired foresight is needed, as we often in

fact build them for distant generations, for prospective commerce, and for the maintenance of peace when wars of the future threaten."

With regard to speed:

"Freight delivered in Philadelphia at 5:00 P. M. to the small boats now plying the canal is landed at Baltimore at 7:00 o'clock in the morning, while the average time for freight delivery by rail is something like two to three days."*

United States Senator P. C. Knox, in an address made some years ago on transportation, referring particularly to waterways, said:

"No subject of national policy has been more distorted by partial views, more disfigured by misapprehension, or more dwarfed by the conflict of local interests, than the Governmental work of improving our harbors and waterways, and yet upon no other one factor does the future expansion of commerce so largely depend.

"The duty of the Government to raise its waterways and harbors to their utmost efficiency was determined long ago by the action of the Government itself. . .

"But it is capable of proof that the radical cheapening of transportation by these projects holds out vast returns to this Government. Merely as a fiscal proposition, it is the best investment of Government funds that can be made. . .

"The relief of the railroads by transferring a good share of their low-class freights to the waterways is not adverse to their permanent prosperity. . .

"When a proposition to spend $40,000,000 on the waterway from Paris to Rouen was submitted to vote, out of a total of 345,000 only 13 votes were in the negative."

The Department of Commerce, Washington, D. C.:

"It must be evident to reflecting men that nothing which advances the interests of the country as a whole can be permanently hurtful to the great transportation systems of the land. There is no reasonable basis for antagonism between the railway and the waterway. Each is the servant of the other, and the success of each is . . . helpful to the other. It is not to the final and the largest interests of the railway that the waterway should be neglected. Each has its own place in the National economy, and the highest success of each depends in no small measure upon the success of the other. It is at this time a matter of National duty to develop the interior waterway, and to give it that part of the Nation's economic life to which its extent and variety entitle it,

*This refers to the Chesapeake and Delaware Canal.

and this should be done as promptly and as thoroughly as possible, by temporary means if need be, in order to get the traffic moving, and then by permanent means in order to make the movement a solid part of our National life."

Many more interesting quotations on this subject could be given, strongly setting forth transportation matters, but the discussion is already longer than intended. The author touches upon a number of features that are of importance and could be discussed at great length and with material benefit.

While a detailed physical analysis and full description of the canal have not been made in all parts, it is hoped that sufficient has been said to show both its feasibility and desirability to serve not only this immediate district, but far beyond into the surrounding country from New York to the Great Lakes and on down to the Gulf of Mexico.

AUTHOR'S CLOSURE: The author of the paper has read and carefully re-read Mr. Lehman's discussion of it, amongst other reasons, for the purpose of ascertaining:

First, whether any of the ten great waterway engineers who are said by the president of the Canal Board to "have thoroughly investigated the question of the feasibility of the canal" have in their reports approved the estimated cost of $65 000 000 for the canal without the Warren and New Castle branches or $72 000 000 including them.

Second, whether his assertion that it is impossible to predict unit costs of transportation, before the canal was built and in operation long enough to determine actual costs, was controverted.

The author has failed to find in the discussion any statement whatever, that the ten waterway engineers have concurred in, or in any way approved the estimate of cost given in the report of the Canal Board, filed with the Governor of Pennsylvania, June 28, 1917. The author ventures to suggest that perhaps the reason is, that the president of the Canal Board in his address before the Chamber of Commerce on February 13, 1917, in referring to these engineers said "The question of the feasibility of the canal including its water supply which is the question to which they have all addressed themselves" and "that it will have

plenty of water and presents no insurmountable engineering problems" is an indication, if not proof positive, that the water-supply was the main question they considered and that these engineers have not approved the estimate of cost.

With regard to the author's assertion that the unit costs of transportation cannot be predicted before the canal has been in operation long enough to actually ascertain them, he would call attention to the fact that Mr. Lehman practically admits its truth when he says, *"It is not possible to ascertain, in advance, the exact character or amount of the tonnage that would be transported, either as to through movement or between intermediate points"*.

The author believes that an impartial reader will agree that the above sentence, quoted from Mr. Lehman's discussion, means exactly the same as the assertion the author made in his paper that "it is impossible for any one to predict, before the canal is in operation, for what part of its round trip a boat will have a load of either coal or ore, or for what portion of the trip the boat will be carrying no cargo at all".

The author believes, if it is agreed that both these statements mean the same, the author's conclusion to his statement that "It is certain, however, that unless every boat carries a full load of ore to the same destination on every trip . . . and waits the same length of time for its return load of coal, the cost per ton-mile will . . . differ with every variation in the trips", is also the logical conclusion of Mr. Lehman's own statement quoted above, notwithstanding the conclusions drawn from the four hypothetical 138-mile trips, two of tow-boat with barges, and two of self-propelled barges, from Braddock to the mouth of Indian Creek for 224 actual working days. The author would call attention to the fact that the old Erie Canal for 12 years, from 1902 to 1913, inclusive, averaged only 204 days open for navigation and it would not be improbable that the same might be the result with the proposed Lake Erie and Ohio River Canal—instead of 224 days.

Mr. Lehman also indirectly accuses the author of disloyalty to his professional brothers in questioning the canal engineer's estimate of cost. In reply he would call attention to the fact that, four of the great canals of the world—the Manchester, Panama, Suez, and Chicago Drainage—actually cost from 100 to 230 per

cent. more than the preliminary estimates, and that the Erie Barge Canal which, so far as terminals in the various towns along its route are concerned, is not yet finished, has already cost about 70 per cent. more than the original estimate.

In view of the above stated facts in connection with these five canals, the author does not hesitate to say that he believes any one of the ten waterway engineers, who have said that the proposed Lake Erie Canal is feasible, would, if he was responsible for its construction, add at least 50 per cent., for contingencies, to the $65 000 000 which is the estimate of cost given in the report of the Canal Board, not including the Warren and New Castle branches, or to the $72 000 000 given as the cost if these branches are to be constructed.

Mr. Lehman practically confesses that the $65 000 000 does not cover the entire cost, when he admits that it does not include interest (he does not say anything as to sinking-fund charges) during construction, and dismisses this very important question of actual final cost by saying, "It will not be as high as thought by the author".

Mr. Lehman also states that "great quantities of coal constantly go by water to manufacturing plants" and that "there would undoubtedly develop a large water-borne coal traffic to works on the canalized portion of the canal alone". The local coal traffic on the Monongahela and Ohio Rivers is for fuel and coal that is to be used in coke-ovens, and the breakage caused by handling is an advantage in the coking process rather than a detriment, and has no bearing on what the canal would carry. As to coal that might go to points on the canal, he evidently has overlooked what the author said in his paper regarding the boats which carry coal to the valley districts going back empty for their return load of coal.

Neither does he say anything as to the cost of coal traffic—on the one Ohio and three West Virginia canalized rivers, whose depths are 4, 4½, 6 and 6½ feet, respectively—destined to go by the Ohio River and a 12-foot canal to Lake Erie.

The author would also say that Mr. Lehmann does not seem to have attempted to refute what the author said regarding his analysis of how the ore and coal traffic might have been carried

on in 1913, the year in which occurred the greatest shipment of coal to Lake Erie from the Pittsburgh coal district.

Finally, Mr. Lehman gives figures as to the unmined coal in West Virginia and Pennsylvania on the Monongahela and Allegheny Rivers, and says that the coal on the former river in Pennsylvania, up as far as to the boundary between the two states, would be enough to furnish the canal with 15 000 000 tons annually for 95 years. Even if his figures are correct he has not attempted to controvert Mr. S. A. Taylor's statement that the loss in value of the coal "would compel the coal companies to continue the use of railroads for lake coal, *even should the canal haul the coal at cost, or even for nothing.*"

CANTONMENT CONSTRUCTION

By Morris Knowles*

OUTLINE

1. General and Organization Problems
2. Topographic Work
3. Temporary Water-Supply, Sanitation, and Disposal of Wastes During Construction
4. Group Lay-Out and Rearrangement of Units
5. Highway, Railroad and Transportation Systems
6. Building Construction
7. Electric Light and Power System
8. Water-Supply System
9. Sewerage System and Sewage-Treatment Plant

Preface. On May 18, 1917, the Conscription Bill was passed by Congress. The United States was then placed face to face with a realization, which but few had understood with the declaration of war, that we must now provide not only for the raising of the necessary armies to be sent abroad, but for housing, provisioning, clothing, equipment and transportation of these armies. The talk this evening relates to the first item of this group—namely, the housing and providing for the living conditions of the soldiers during their early training. This, of course, means not only places in which to sleep and to eat, and facilities for training, drilling and recreation; but also for all of the public utilities required to furnish transportation, light, heat, water, and disposal of wastes for cities of no mean size averaging 35 000 to 40 000 soldiers.

The various divisions of the War Department and civilian committees aiding the Council of National Defense had for some time been engaged upon a consideration of these problems. The

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attention given directly to the subject which interests us this evening was by a subcommittee of the War Munitions Board, of the Advisory Committee to the Council of National Defense, which subcommittee was entitled the "Emergency Engineering and Construction Committee." Studies had been made, in conjunction with social and housing experts, town planners and engineers, as to the proper grouping, capacity of buildings, air-space and other miscellaneous arrangements.

First and foremost was the consideration given to the question of whether soldiers should be housed in wooden buildings—i. e., in cantonments—or in tent camps. It was finally decided—on account of the lack of time, scarcity of material and the difficulty, therefore, of securing sufficient lumber—that all of the National Guard units, comprising sixteen divisions, should be housed in the South, using the tent material which they already largely had. The other sixteen divisions, provided for in the first army of the selected youth of the Nation, were planned to be housed in sixteen cantonments which, generally speaking, were placed in the northern part of the United States.

The tent camps provided for but few buildings—namely, mess halls, latrines, showers, storehouses, and some miscellaneous assembly, recreation, and office buildings. We had the good fortune—good fortune because of the added unique experience derived—to be placed in charge as Supervising Engineers of one of these camps—namely, Camp McClellan, about six miles north of Anniston, Ala., used for the National Guards of New Jersey, Delaware, Maryland, the District of Columbia, and Virginia. Some of the data herein presented, and certain pictures, will relate to this construction.

In the cantonments, not only were all of the buildings above mentioned constructed of wood; but barracks for housing soldiers, and sheds for horses and equipment were also built of wood.

The average annual temperature at Camp Meade is 58 degrees. The average year contains 131 cloudless days and 124 partly cloudy. Camp McClellan has an average annual temperature of 61 degrees. The highest recorded temperature is 153 degrees, with an average of 208 days of sunshine during the year.

The barracks, as originally planned for 200 men, were to be

43 feet wide, 140 feet long, and two stories high. Later, as will be explained, other buildings, for 66-man units, were planned two stories high and 30 by 60 feet in size. Most of the information presented this evening with regard to cantonments relates to Camp Meade, Maryland, where we were Supervising Engineers of the construction. This cantonment, located about 25 miles north of Washington, 15 miles north of Baltimore and 14 miles from Annapolis, originally planned for the troops of Pennsylvania, was finally utilized for those of the eastern part of the state, and also those of the states of Maryland and Delaware and the District of Columbia.

1. *General and Organization Problems.* There was early organized in the Quartermaster's Department, under the supervision of Brigadier General (then Colonel) I. W. Littell, a section called the Cantonment Division, to have charge of all of the design and construction of these new units for the War Department. General Littell still remained in charge of what is known as the Construction and Repair Division, having to do with all of the construction and maintenance of regular army posts situated all over the country.

The instructions to Colonel Littell might well have been paraphrased into the following apt quotation from the Nov.-Dec., 1917, number (v. 32, p. 439) of the National Geographic Magazine:

> "You are to provide all of the mess halls and other general buildings for all of the 16 National Guard mobilization camps. And while you are doing that you will not forget your regular work of expanding and keeping in repair the housing facilities of the Regular Army posts. . .
>
> "Yes, I know it is a large order—in fact, a tremendous proposition—but these are tremendous times, and I'll have to ask you to execute it within four months. Of course, I realize that you will, in its execution, spend the money three times as fast as the world mines its gold, but at the same time I expect you to render an account which will show that every penny has borne an honest burden."

The Cantonment Division not only had the advice of a number of loyal engineers, architects and civicists in planning for this work, but also perfected a complete organization, dividing the labors into four different groups, reporting through Major (then Captain) W. H. Oury, and Major (then Captain) R. C. Marshall, both of the Regular Army.

The four groups were composed as follows: One having charge of design was under the direction of Major Frank N. Gunby, formerly associated with Mr. Charles T. Main of Boston, President of the American Society of Mechanical Engineers. He has been assisted by Major F. B. Wheaton on building construction; Captain L. C. Doten on sewerage and highways; Major Dabney H. Maury of Chicago; Clarence Goldsmith and W. M. Johnson, of the National Board of Fire Underwriters, on water-supply systems; and Captain George Gibbs, Jr., of Boston, as chief draftsman. The second division was in charge of Major Robert E. Hamilton—formerly purchasing agent of Stone & Webster, of Boston, where he had charge of ordering and transportation of materials—assisted by a group of deputies. Later, this work was placed in charge of Major Willcutt who, during the construction of the National Guard camps, had general charge of these. The third division was in charge of Major M. J. Whitson of Seattle, Washington—formerly partner of Grant, Smith & Company, contractors. Major Whitson had charge of construction, assisted by a staff of division engineers—Major Peter Junkersfelt of Chicago; Major Philander Betts of Newark, N. J.; Major Ezra B. Whitman of Baltimore; and Captain Kebbon of Boston. The fourth division was under Major W. A. Dempsey, who had supervision of all accounts and financial records, and control of all field auditor's work, and Major Evan Shelby who advised as to all forms of contracts and legal questions.

The stupendous nature of the task ahead can be appreciated if you will remember that the entire program called for the construction of some 26 500 buildings, to house 675 000 men of the National Army; two embarkation camps, for 43 000 men; a quartermaster's training camp, for 18 000 men; repair shop units and ordnance depots; also the necessary lay-out and buildings for sixteen National Guard camps, to care for 620 000 men—and for accomplishing this so that troops could be housed in the early autumn. Where in the world's history can you find a similar demonstration of quickly energized co-operation and utilization of the Nation's resources to be effective with so little preliminary planning. Just think of the expenditure of

$187 000 000 in six months—three times the maximum rate of expenditure on the Panama Canal during its construction—increasing many fold the normal building and transportation activities of the country, without substantial increase in rates of pay or cost of materials. That it was possible is due in no small measure to the already appreciative consideration, of the various departments of the army, of the needs in this respect; but also to the willing and loyal sacrifice, and voluntary service—in many cases without any compensation whatsoever—of patriotic citizens; serving as administrators, advisors and designers to the departments and the committees in Washington.

With the general design and central management of construction under care of such able men as above mentioned, the other important portion of the work related to the choosing of responsible men to have care of the construction at each individual camp and cantonment. This was the work of the previously mentioned very important subcommittee on Emergency Construction and Engineering, a part of the War Munitions Board of the Council of National Defense. This committee was under the able leadership of Mr. William A. Starrett, architect and builder, formerly of the Thompson-Starrett Co. His committee selected the contractors and supervisors of construction for the various encampments. Contractors did not bid competitively for the work but were chosen from those who had shown by equipment and execution their ability to handle work of such magnitude and upon such a scale. The form of contract for the construction, while it provided for a cost-plus-percentage method of payment, did not indicate a profit as high as 10 per cent.—as has frequently been stated—but was on a sliding scale, decreasing with the gross cost and depending upon the size of the various contracts; and the maximum contractor's fee for the entire construction of a cantonment was limited to a lump sum of $250 000 which, when overhead expenses and interest charges are deducted, would leave a net profit to the contractor of something less than two per cent.

This committee also, with the advice and help of members and committees of the various engineering societies, determined upon forms of contract to be used with contractors, and also with

engineers who were later entitled Supervising Engineers of Construction. The recommendations of this committee as to the selection of contractors and supervising engineers, after being passed upon by the War Munitions Board, were submitted to General Littell and the Secretary of War, who made the final appointments—as also those for constructing quartermasters. The latter were chosen from the ranks of the Quartermaster's Department of the Army—generally being Majors, either of the Regular Army or the Reserve Corps—and, in case of some of the National Guard camps, from the Quartermaster's Department of some of the National Guards.

The 16 National Army cantonment sites were selected and approved during the month of June; contracts executed between the middle and the end of that month, and work begun between the middle of June and the first week of July. On September 5, or in less than three months, more than 300 000 men could have been taken care of, and on the first of December—notwithstanding the great increase in construction involved by prescribing additional air-space per man in the barracks, by material changes in army organization and by the addition of several units; such as provision for training battalions, base hospitals, horse remount and training depots—such energy had been devoted to the prosecution of the work that cantonments with all additions were practically completed for the entire 650 000 men.

Figures which will illustrate the magnitude of the work may not be amiss. The cantonments themselves called for about 800 000 000 feet of lumber, requiring for delivery 37 000 cars; and 40 000 cars of other material were also needed. For all the camp, cantonment and embarkation station construction, there were required approximately 112 000 cars, 172 000 doors, 112 000 kegs of nails, 314 000 barrels of cement, and 282 miles of wood and cast-iron waterpipe. For the equipment of the 16 cantonments, 350 carloads of cooking ranges and 2500 carloads (40 000 stoves) for heating, were required. Steam heating in addition to that for base hospitals and officers' quarters was not provided except in four cantonments, in the northerly part of the country, where regimental heating-plants were used.

Construction plans of all buildings, and plans of regimental and other organization lay-outs were furnished from the Washington office and it became the duty of the construction quartermaster and the supervising engineer to rearrange the grouping in such a way that, while adhering to the military requirements and necessities, buildings could be placed without excessive grading or great expense due to foundations; and at the same time to provide adequately and conveniently for the highway systems, water-supply and distribution, sewerage and drainage.

For the purpose of advising as to general lay-out and arrangement of buildings, various members of the American Institute of Landscape Architects were designated to confer with the constructing quartermaster and the supervising engineer, for the purpose of securing a convenient and well-arranged municipality. These men were called "town planners," and at Camp Meade we had the assistance of Mr. Owen Brainard of the firm of Carrere & Hastings, architects in New York City. The writer has been associated with Mr. Brainard on a number of projects and can testify to his admirable judgment and comprehensive knowledge. The convenience and attractiveness of the lay-out plan at Camp Meade and the accessibility of the highway transportation system are due in no small measure to Mr. Brainard's excellent capacity in dealing with such problems.

The organization for the construction of a cantonment provided not only for the general contractor—with the necessary subcontractors under him for special purposes, such as electrical work, water filtration, plumbing installation and sewage treatment; and in some cases for forest cutting and clearing—but also another contractor for highway construction, and occasionally special contractors for special community buildings. The general contractor at Camp Meade was Smith, Hauser & McIsaac—a concern of great experience and capacity doing work all over the United States. The electrical work was sublet to Riggs, Distler & Stringer, of Baltimore. Claiborne & Johnson of Baltimore and, later, Mr. William B. Shaffer of the same city, were separate contractors on highways, gravel and concrete. The work of clearing and forest cutting was done by Beaseman & Company, of Baltimore, for the state of Maryland, which agreed to furnish

the cleared site to the Government. This work—for the state—was under the particular direction of Major W. W. Crosby, M. N. G., later Lieutenant Colonel of the 304th Engineers, Twenty-ninth Division, U. S. A., at Camp McClellan. The constructing quartermaster was Major Ralph L. Procter, U. S. R. Q. C., of Baltimore; a man of long experience in general construction work, whose indefatigable attention to detail was marveled at by all.

The force of the constructing quartermaster included several military assignments for care of various portions of the work and the military detail; also several civilian assistants. Of the several assistants, Major Keith Compton and Mr. W. W. Pagon of Baltimore rendered loyal service in an engineering capacity. Mr. H. F. Fairbanks, assigned for the Office of Public Roads, of the Department of Agriculture, advised with regard to highway construction. The accounting and inspection work was done by the force of the field auditor, so called. The men for this work were chosen from the National Association of Certified Accountants. They took charge of all bookkeeping and time accounts, and also the records of receipt, inspection and character of all material. Mr. E. L. Hatter, of Baltimore, was the man selected for Camp Meade, and to his breadth of experience and good judgment is due much of the success in carrying out this difficult and sometimes annoying piece of work on all construction.

At Camp McClellan the entire direction of the work, detailed as well as general, was under Colonel (then Major) Charles L. Dulin, Mississippi National Guard, whose extensive experience in matters of this sort, familiarity with the organization needed for handling such work, and positive and directive nature, made for the overcoming of obstacles that seemed almost insurmountable. The general contractor was John G. Chisholm & Company, of New Orleans—of considerable experience in the South—who had previously built a machine-gun camp of smaller size on the same government reservation nearer Anniston.

It would be impossible for the supervising engineer to give credit to all who loyally assisted in this important and trying work, but it is interesting and seems fitting to mention some of the Pittsburghers. Except for the untiring efforts of all the

force, who so faithfully gave of their whole energy, the success of these great operations would not have been possible. The Pittsburgh office was used as a nucleus for the building up of the organizations of both Camp Meade and Camp McClellan, but important positions were filled in several cases by recruits drawn from friendly agencies, consisting of municipalities and corporations all over the country. At Camp Meade the principal division engineer was Mr. C. M. Reppert, employed in the Bureau of Engineering, Pittsburgh. The work was then divided into three groups. One was under Mr. Leo Hudson of Pittsburgh, who had to do with general lay-out, staking of buildings, design and supervision of construction of sewer systems and highways. Under him served Mr. C. F. Brown of the Bureau of Engineering, Pittsburgh, as principal assistant; and Mr. Nathan Schein, of the Bureau of Engineering, Pittsburgh; Mr. M. L. Schmidt, of the county engineer's office; and Mr. S. A. Smith, of the Bureau of Water, Pittsburgh. Another division was under the head of Mr. William H. C. Ramsey, of York, Pennsylvania, who has had extended experience in waterworks construction and management. He had charge of waterworks and electric systems—both design and construction—and under him, as principal assistant, was Mr. Elwood Avery who later served in the same capacity in charge of water-supply work at Camp McClellan. There were also employed Mr. William L. Hirsch and Mr. William Donnan, both of the Bureau of Water, Pittsburgh. Mr. A. B. Morrill, of the Pittsburgh office, also served on the water-supply work here and, later, on the water-supply work at Camp McClellan.

The third division had to do with headquarters, office work and general management. This was under the charge of Mr. John D. Stevenson, assistant engineer of the Bureau of Engineering, Pittsburgh. The drafting-room was under the charge of Mr. Joseph Lambie of the faculty of the School of Engineering, University of Pittsburgh. All adaptations, transformations and new designs, as well as progress sheets and record plans were prepared in the local camp office. The matter of camp sanitation and temporary water-supply, also in the headquarters division, was under the direction of Mr. Francis C. Foote of the Pitts-

burgh office, who later, as Second Lieutenant in the Engineer Corps was assigned to Camp Dix. Mr. W. W. Daly of the office of Chester & Fleming, of Pittsburgh, succeeded Mr. Foote. The office management, consisting of bookkeeping, accounting, stenographic force, and filing, was under the charge of Mr. A. L. Jacobs of Main, Squires & Company, of Pittsburgh, which concern has been associated with Morris Knowles, Inc., on many important pieces of work, and gave us loyal service not only at Camp Meade, through two assistants (the second was Mr. Carl Simpson), but also at Camp McClellan in like capacity through the services of Messrs. James H. Young, Jr., and W. R. Cavenney. Camp Meade men will never overlook an occasion of the mess and quarters organization, also under the headquarters organization, and under the charge of Mr. William R. Conrad. His ever-ready willingness and unruffled good nature did much to make for a contented and happy force.

The organization at Camp McClellan was under the immediate direction of Mr. Maurice R. Scharff, principal assistant engineer of the Pittsburgh office. Assisting him was Mr. A. B. Hargis, of the city engineer's office, Birmingham, Alabama, in charge of headquarters division, office work, etc., who later became principal division engineer upon the departure of Lieutenant Scharff to France to serve in the Engineer Corps. The general field work and lay-out of units and buildings was under the charge of Mr. R. L. Totten, who had for some time been in private engineering practice in Birmingham, Alabama. He later became superintendent for the contractor. The water-supply work was under the charge of Prof. C. J. Davis, Jr., of the University of Alabama, at Tuscaloosa; and the electrical work under the direction of Prof. C. F. Wittig of the same institution. The temporary water-supply and camp sanitation work was under the charge of Mr. C. C. Hommon of the city sewage disposal works of Atlanta, Georgia—later Commander or Captain in the Sanitary Corps of the U. S. A. Medical Corps and assigned to Camp Gordon—assisted by Mr. R. D. Bates of the Pittsburgh office. Later, the work was under the sole direction of Mr. Bates, who had seen important similar service in Servia with the Red Cross during the early months of the present war.

It may appear to some that the multitude of counsel, the ramification of committee assignment, and the mixture of civilians with those of military training, in Washington and elsewhere, would lead to confusion, criticism and lack of co-ordination. It is fair to say that in some cases this was apparent to a slight extent; but when one thinks of the vast magnitude and varied nature of the war preparations being undertaken simultaneously by our Government, the surprising thing is that there should be so little of confusion and so little of the lack of harmony. And the astonishment of the world will undoubtedly be manifest when it is realized that the democracy of the United States has so risen to the opportunity, and prepared for a war duty so out of keeping with its usual functions. We may, indeed, be gratified that there has been so little opportunity for criticism due to misplaced confidence, lack of efficiency, or intent to confuse or confound. It is really pleasing—and I am glad to testify—that we have found that men from various walks of civil life can be gathered together abruptly with the complete severance, for the time being, of the usual ties of home and pleasure; without comfort and convenience, and with everyone devotedly applying himself to the job in hand, namely, of contributing all energy and equipment to the completion of this necessary detail in winning this war in which we are now engaged. All are glad, I am sure, who have served in this humble capacity and have played a part in that which will go down in history as one of the several great achievements of our Nation, suddenly thrust into this international maelstrom.

The duties and authority of the supervising engineer covered all the work herein mentioned, beginning with the preliminary surveying and topography; lay-out of units and buildings; administration of temporary water-supply and sanitation; adaptation of general plans and designs for all utilities—such as highway, water and electrical systems, and supervision and inspection of construction—except that, at Camp Meade his jurisdiction stopped at a point five feet from the buildings; though at Camp McClellan his force also looked after building construction and equipment.

2. *Topographic Work.* One of the first requisites, which was not, however, fully appreciated by everyone, was that there should be an accurate topographic map of the site upon which it was proposed to build a cantonment. In general, for most places, the maps of the United States Geological Survey existed, these being on the scale of about one mile to the inch and with contour intervals of 20 feet. While some preliminary studies could be, and were made from these maps or their enlargements, it was quite difficult to make the required detail location; especially to determine the minor grouping of buildings and the arrangement necessary to avoid excessive grading, or raising of buildings at one end. For this reason prompt preparations were made for the making of topographic maps at both Camp Meade and Camp McClellan.

Fortunately we were able to make excellent arrangements with the well-known firm of topographic engineers: Brown and Clarkson, of Washington. The promptness of response and the rapidity and excellence of execution of this force of men, under the personal direction of Mr. William M. Brown, contributed in no small degree to the success of the undertaking and permitted us to proceed rapidly with the location and determination of the grades of units. This work was done on plane-table sheets and as many as three parties were operating at one time. First, of course, it was necessary to establish a controlled co-ordinate system, which was later extended and tied in with the controlled traverses of the entire area. The plane-table maps were made on the scale of 200 feet to the inch, with five-foot contour intervals; each sheet covering 4000 by 5000 feet, or 275 acres. By July 14, a total of 3000 acres had been thus mapped and plotted at Camp Meade; which was sufficient, with a few minor later additions, to lay out all the units of the entire cantonment.

A word as to the excellence of the plane-table work may not be amiss. Our own local experience would have led to us to carry on this work by stadia, in the customary way with which so many of us are familiar; but if we had any misgivings at all as to the advantages of plane-table work, they soon disappeared with the very general success which we attained at both Camp Meade and Camp McClellan. It is our opinion that in no other

way could the work have been done so quickly and satisfactorily as by the plane-table method.

As an example of unusual rapidity, let me relate that, due to the exigencies of the occasion, it was necessary for various reasons at Camp McClellan to determine promptly what could be done with a certain portion of the entire reservation, and whether there would be room within certain foot-hills to place a division of troops. It was therefore decided to make a tentative topographic map of about 2500 acres, on a scale of one inch equal to 800 feet, with contour intervals of 10 feet. It was determined only Thursday night that this was necessary, and it was required that the completed map be available by Sunday night. By the use of a horse and buggy for the plane-table man and a recorder, with two rodmen on horseback, the entire field work was executed in two days and the map was ready for study on the evening of the third day. Its handy sizes, 15 by 21 inches, made it extremely valuable and convenient, and its accuracy was favorable when compared with the larger and more complete 200-foot scale map which was later executed.

As a necessary part of the preliminary topographic work—and particularly to permit of construction—it was required that woods and underbrush should be cut out, swamps drained and, in several cases, minor streams diverted. It was the endeavor in this clearing to save all the good timber, not interfering with the location of buildings or highways and not on land required for drill ground immediately adjoining the area of the regiments. Some considerable success was attained but, unfortunately, it is probable that many valuable stands were removed through the energetic desire to register promptness in execution. A complete forestry map was made of all standing growth at Camp Meade in the area allotted to building purposes.

3. *Temporary Water-Supply, Sanitation, and Disposal of Wastes During Construction.* The concentration under temporary camp conditions of thousands of workmen and laborers requires the utmost care, in order that the seeding of infectious and contagious diseases shall not occur, due to insanitary conditions.

The first and most important problem is to secure sufficient

good drinking water. At Camp Meade the only sources available were the farm wells, generally shallow, and a few deeper drilled wells located at some of the settlements near-by on the railroad lines. At Anniston, but little water of this sort was obtainable, and that was from hillside springs. It was not until the city supply was piped out from town that there was sufficient water for the construction force. Many of the farm wells were quickly drawn dry and most of them did not have a satisfactory supply from the standpoint of purity. The care of this work at Camp Meade was under the supervision of a force from the supervising engineer's office, directed and guided by the officer assigned to the camps by the Medical Corps of the Regular Army. At Camp McClellan, during the construction period, the entire operation of such detail was under the control of the supervising engineer's office, acting under the authority of the constructing quartermaster.

It was necessary to station men at the various wells which were available. These men were employed to check up the filling of tank carts and automobile trucks; to chlorinate water where necessary; and to report any infractions of the general scheme for procuring and conserving a good supply. Military detail was required in several instances. It was difficult, as has always been the experience, to prevent people securing water from near-by sources, whether polluted or not; and to secure complete observance of sanitary precautions, unless the complete program was carried out and enforced by the men of our own force. Frequent bacteriological analyses were made of all sources of supplies and changes in use were made according to requirements and conditions.

As the contractor's force was expanded and the construction area extended, the requirements of securing and distributing the necessary water-supply were gradually increased; passing through the stage just described into one where driven wells were utilized as sources of a greater supply, and one where a temporary, and finally the permanent piping system was used for distribution. This involved the setting up and operation of many small pumping units, storage and distribution tanks, facilities for properly sterilizing the supply, etc.

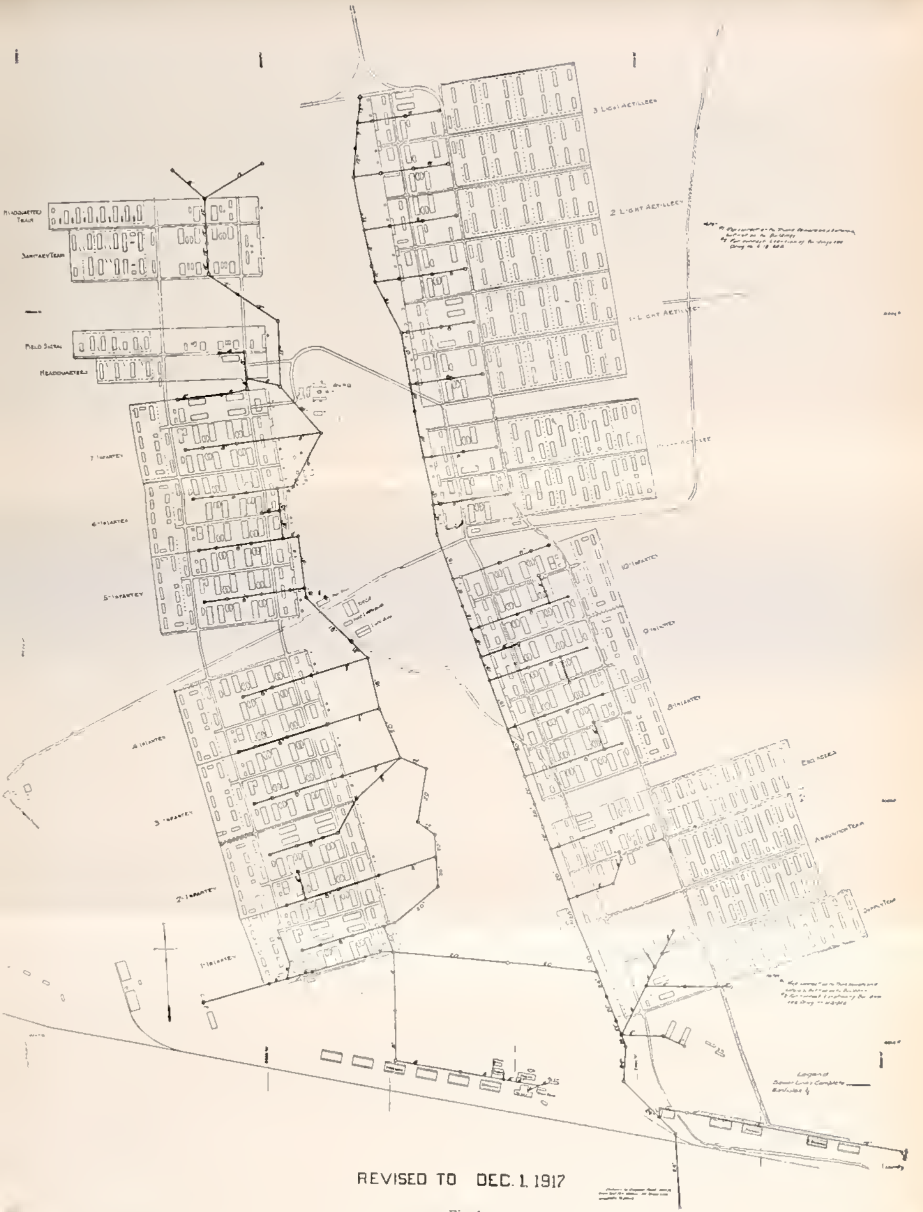
The problem of obtaining a prompt and satisfactory supply, under the trying conditions of caring for many thousands of men quickly transported to a site, is a most serious and annoying one: but that we were successful in maintaining regulations and requirements in this, as in other forms of sanitation, is evidenced by the fact that during the entire construction period, with a maximum of over 10 000 men employed, there were no cases of infectious diseases, like typhoid and malaria fevers; only one death occurring and that due to accident. And since completion Camp Meade, for the 79th National Army Division, has generally ranked lowest in morbidity and mortality rates, the former rate having been as low as 6.5 cases of illness per 1000, generally averaging from 10 to 20, while the average for all the National Army cantonments has ranged from 25 to 50. Similar comparative and satisfactory results were also achieved at Camp McClellan.

The disposal of human excrement was in general in accordance with United States Army regulations, modified slightly, due to the lack of military dicipline among the workmen engaged. Latrines were of the box type, fly-proof construction with self-closing lids, and pits dug in the ground from six to ten feet deep, depending upon the nature of the ground. Relocations were made and pits filled from time to time. In some cases spraying with lampblack and oil was sufficient, but generally pits were burned out daily during the period of greatest activity. All seats were scrubbed daily, frequently using some disinfectant. As soon as the sewer system was partially finished, and the trunk lines completed to the site of the treatment unit, the lavatories were placed in use and the effluent, after sterilization, was discharged into the creek near the treatment plant.

The disposal of kitchen waste and refuse was one of the most difficult problems. The plan later evolved of selling garbage to farmers, for feeding hogs, was not available during the early period of construction. The army type of rock-pile incinerator was first used, but was quite unsatisfactory, due to the volume of material and accompanying nuisances, and was soon abandoned. Later modifications—of brick and concrete construction, with grates made of iron, upon which the refuse was placed, and

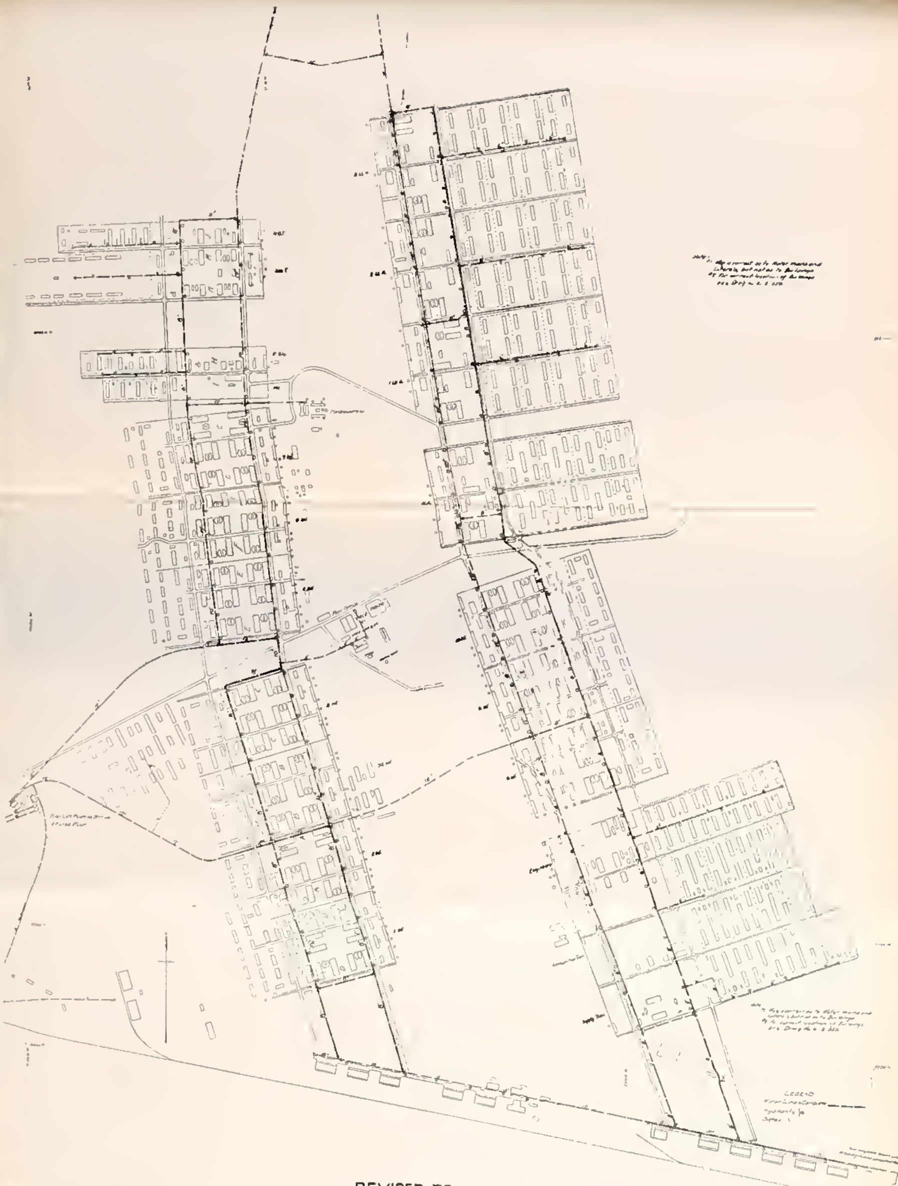
with wood fires burning underneath—were adopted and generally used until the inauguration of permanent camp and cantonment methods. For the National Army, it is planned to sell most of the refuse, and a contract for this amounts to $446 000 for all the National Army cantonments, with an additional amount of $198 000 for the sale of manure. Small incinerating plants, however, have been built at each cantonment, where material will be sorted, and that which is not taken away will be burned. Company incinerators, of pan type on brick walls, using wood fires, are used at National Guard camps.

4. *Group Lay-Out and Rearrangement of Units.* A division of troops as originally planned comprised four brigades, three of which were infantry of three regiments each, with one additional regiment of infantry to take the place of the discarded cavalry unit. There was also one brigade of three regiments of light artillery, with one regiment of heavy artillery; and a regiment of engineers, with engineers' train, supply train, ammunition train, sanitary train, division headquarters unit and headquarters train. If laid out in a straight line such a division would occupy a space from three to four miles long and about a mile wide. If placed in the form of the letter "U," the division would occupy an area about two to three miles long and one to one and one-half miles wide, depending upon the contour and shape of the ground. A general plan of the Camp Meade lay-out is shown in Fig. 1–2. Fig. 1 shows the sewer system, while Fig. 2 shows the water-supply system.



Headquarters Train
Sanitary Train
Field Signal
Headquarters
3 Light Artillery
2 Light Artillery
1 Light Artillery
7 Infantry
6 Infantry
5 Infantry
4 Infantry
3 Infantry
2 Infantry
1 Infantry
10 Infantry
9 Infantry
8 Infantry
Engineers
Ammunition Train
Supply Train
Legend
Sewer Lines Complete
REVISED TO DEC. 1, 1917

Fig. 1.



REVISED TO DEC. 1, 1917

Fig. 2.

Given the typical grouping of a division and the topographic data, it was then necessary to make the lay-out for the particular place; and, as before stated, it was decided to place this in the form of the letter "U," one leg being upon the east side and the other upon the west side of the Rogue's Harbor branch.

As topographic records were not immediately available to lay out the whole of the arrangement in detail, the first problem was to see whether we could start upon a brigade, or three regiments of infantry as originally planned, with the belief that the rest of the division would fit in harmoniously with this unit as thus fixed. It was thus determined that three regiments of infantry could be laid out on the west side of the stream, and south of the Severn-Portland road. It was later found that an additional or odd regiment of infantry could be placed south of this, and the remainder of the units were then grouped according to the general plans shown. This enabled us to place the divisional storehouses adjoining and paralleling the railroad track, along the southerly boundary of the building area. The units added later—namely, the base hospital, the training battalions and the horse remount depot—were placed without serious inconvenience or disarrangement of the general scheme.

The general arrangement of infantry and artillery regiments is as shown in Fig. 1. Most of the units were laid out in accordance with the general scheme furnished by the Washington office. In many instances it was extremely troublesome and expensive to place all buildings in a regimental unit exactly according to the typical plan, and it therefore became necessary, on account of the configuration of the ground available, to rearrange units; placing some buildings differently—all for the purpose of avoiding excessive time and expense in grading for the buildings.

In general this rearrangement was accomplished in groups of two or more buildings, without interfering with the regular general alignment, and without affecting the distance between regimental roads. In some cases, however, due partly to the later authorizations for additional buildings when the size of companies and regiments was changed, some new buildings were required for administrative purposes and it was necessary to make some changes in these respects. Progress drawings were pre-

pared from time to time, and these not only furnished the information for staking out all buildings upon the ground, but were also used for the purpose of construction and, later, as complete record plans when the work was finished.

The problems at the National Guard camp were, of course, not as serious, because the buildings were fewer in number, generally including only the mess hall at the upper end of the company street and the shower and latrine at the lower end. However, in the case of Camp McClellan the topography was studied with great care, and groups were located in such a way that, if it be desired later to change to a cantonment with building construction, this may be done without serious delay or changing of arrangements; and a sewer system may be built with ease and convenient arrangement of grades and trunk lines. (See Fig. 3 for plan of Camp McClellan.)

With the location of units determined according to the general group lay-out, the first problem was to locate the four corners of a regimental unit and, about the same time, also, the center line of the regimental streets. Later the four corners of each building were staked and batters were placed for the buildings. Due to the exigencies of clearing, blowing out stumps and hauling lumber, much of the staking was repeated many times, and in fact instances have been known where a complete staking of a regimental unit was required the fourth time, much to the exasperation of the field-party men and all concerned.

5. *Highway, Railroad and Transportation Systems.* It was originally expected that the highway systems of cantonments would be constructed before other work was begun, in order that transportation could be readily conducted during the construction period. In fact, arrangements for building these highways according to certain specifications of bituminous macadam had already been made. Time, however, did not permit and the exigent nature of the work brought about a situation such that during construction there were available only the ordinary unimproved country roads, and such temporary thoroughfares as were boldly carved out over the farm lands. In addition to the regimental thoroughfares and company streets, shown upon the typi-

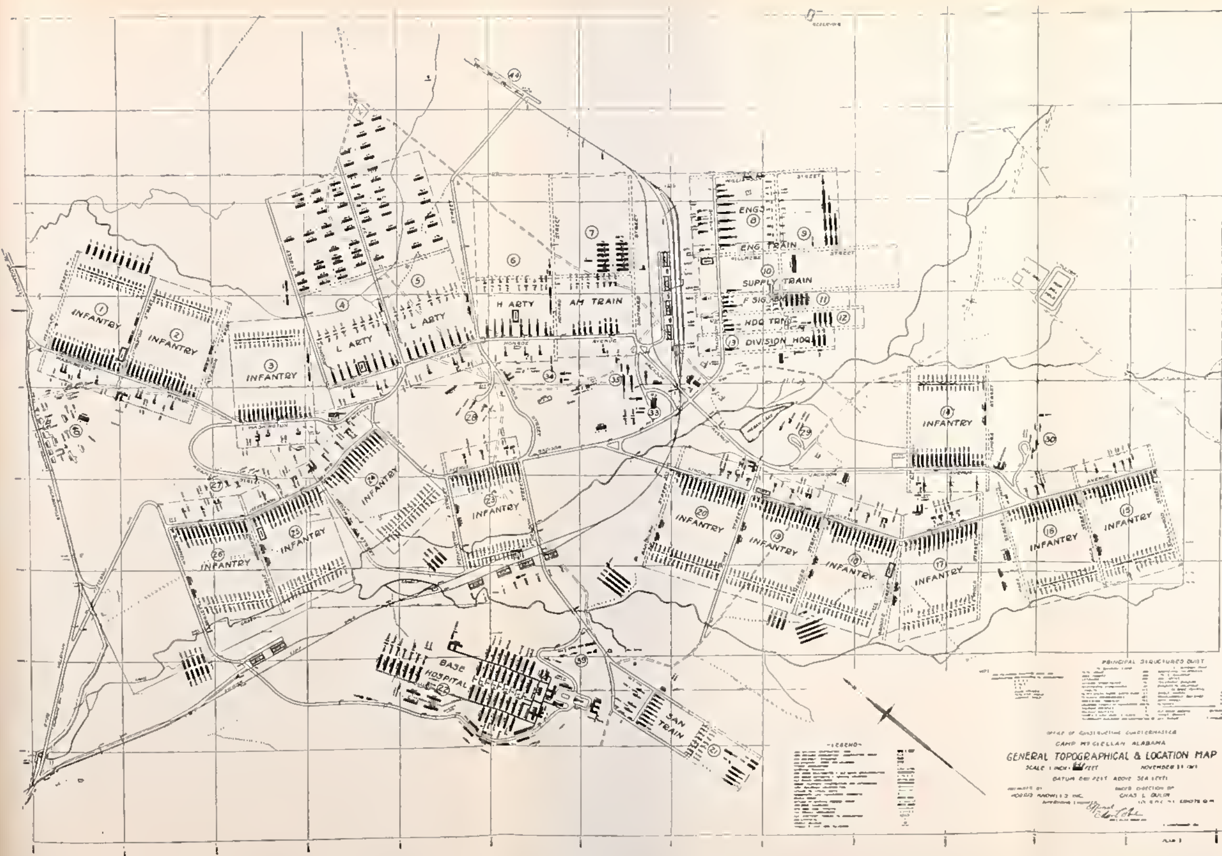

CAMP MCCLELLAN ALABAMA
GENERAL TOPOGRAPHICAL & LOCATION MAP
LEGEND
PRINCIPAL STRUCTURES BUILT
1 INFANTRY
2 INFANTRY
3 INFANTRY
4 L ARTY
5 L ARTY
6 H ARTY
7 AM TRAIN
8 ENGS
9 ENG TRAIN
10 SUPPLY TRAIN
11 F SIG BN
12 HDQ TRN
13 DIVISION HDQ
14 INFANTRY
15 INFANTRY
16 INFANTRY
17 INFANTRY
18 INFANTRY
19 INFANTRY
20 INFANTRY
21 SAN TRAIN
22 BASE HOSPITAL
23 INFANTRY
24 INFANTRY
25 INFANTRY
26 INFANTRY
27
28
29
30
33
34
35
39
44
STREET
AVENUE
MONROE
WASHINGTON
JACKSON
LINCOLN

Fig. 3.

cal plans, various connecting roads and improvements of existing roads, were planned at Camp Meade. The so-called outer-loop road, connecting with the road in front of the divisional storehouse and utilized for heavy traffic, was constructed of concrete, 18 feet in width, and about four miles long. In addition to this a cross-road through the camp—connecting with Annapolis Junction on the B. & O. Railroad via Portland—and a connecting highway to the Severn road, were also built of concrete; a total length of about four miles. The inner-loop road, the road to the hospital, and the connecting road to the Jessup road, were planned to be built of gravel. The total length of these roads is about eight miles. The sole means of transportation into the camp area was the single trolley line of the W. B. & A. Railroad Company. This was once a steam line owned by the B. & O. Railroad, running from Annapolis Junction to Annapolis. It was taken over and operated as an electric line when the W. B. & A. built a line between the two main cities, with a transfer point at Naval Academy Junction. When we first appeared upon this scene, the branch line operated only four cars a day, and it was upon this road that for some time all materials were brought into camp, though in addition, during the beginning, many of the preliminary materials, such as cots, bedding, food supplies and some lumber and contractor's equipment were brought by motor trucks from Baltimore.

It was planned early, but executed none too soon, that the B. & O. Railroad should run in a line parallel with the W. B. & A. from Annapolis Junction to the Disney Switch, so called, with construction tracks and yards at Portland. Similarly, the Pennsylvania ran in from Odenton, as far as the Rogue's Harbor branch, thus supplying yards located at Admiral. At first the freight cars of the two steam roads were operated over the electric line. Later it was decided that the trolley company should run a line for passenger and express service up to the middle of the building area. This was done by running a spur off the main line at Admiral, crossing underneath the W. B. & A. and the steam railroad, and following up the left bank of the Rogue's Harbor branch, terminating in a loop located near the Portland

road and near the civic or community center. Here, there are located not only the post-office, but the central Young Men's Christian Association building, the hostess house of the Young Women's Christian Association, and the central buildings of the Knights of Columbus, and the Young Men's Hebrew Association. The camp theater is also located near this group.

6. *Building Construction.* There are 1514 buildings at Camp Meade, of which over 1400 are in the general arrangement, 57 at the base hospital and 7 at the remount; the others being miscellaneous utility and service buildings. The total amount of lumber required at Camp Meade was 45 000 000 feet board measure. The average daily use was 500 000 feet. A maximum of nearly 1 000 000 feet was placed in buildings in one day; and for half of one month about this amount was hauled each day from the freight yards to the sites of the buildings.

There is much that is interesting in the history of the development of cantonment buildings for barracks. The desires of the house planners and social workers for a large air-space per man, and for the community rooms or commons in each barracks, were not fully realized. Finally, an air-space of 300 cubic feet per man was agreed upon, but this was increased later, when the change came in the size of company and regimental units, to provide for 500 cubic feet of air-space per man in a large barracks occupied by 150 men. This was done so that two companies now occupy three barracks; the mess hall and kitchen of the middle building being converted into a dormitory, with the division through the middle building, and 100 men in each half. Except for certain of the non-commissioned officers there are no provisions in the dormitory buildings for keeping clothes or other personal property. Such articles are hung on nails from the posts or rafters, or kept in boxes or cases under the cots. The dormitory rooms are large open rooms with cots arranged throughout the entire floor area. Double-deckers were at one time suggested but later discarded. (A view of these buildings is shown in Fig. 4–5.)

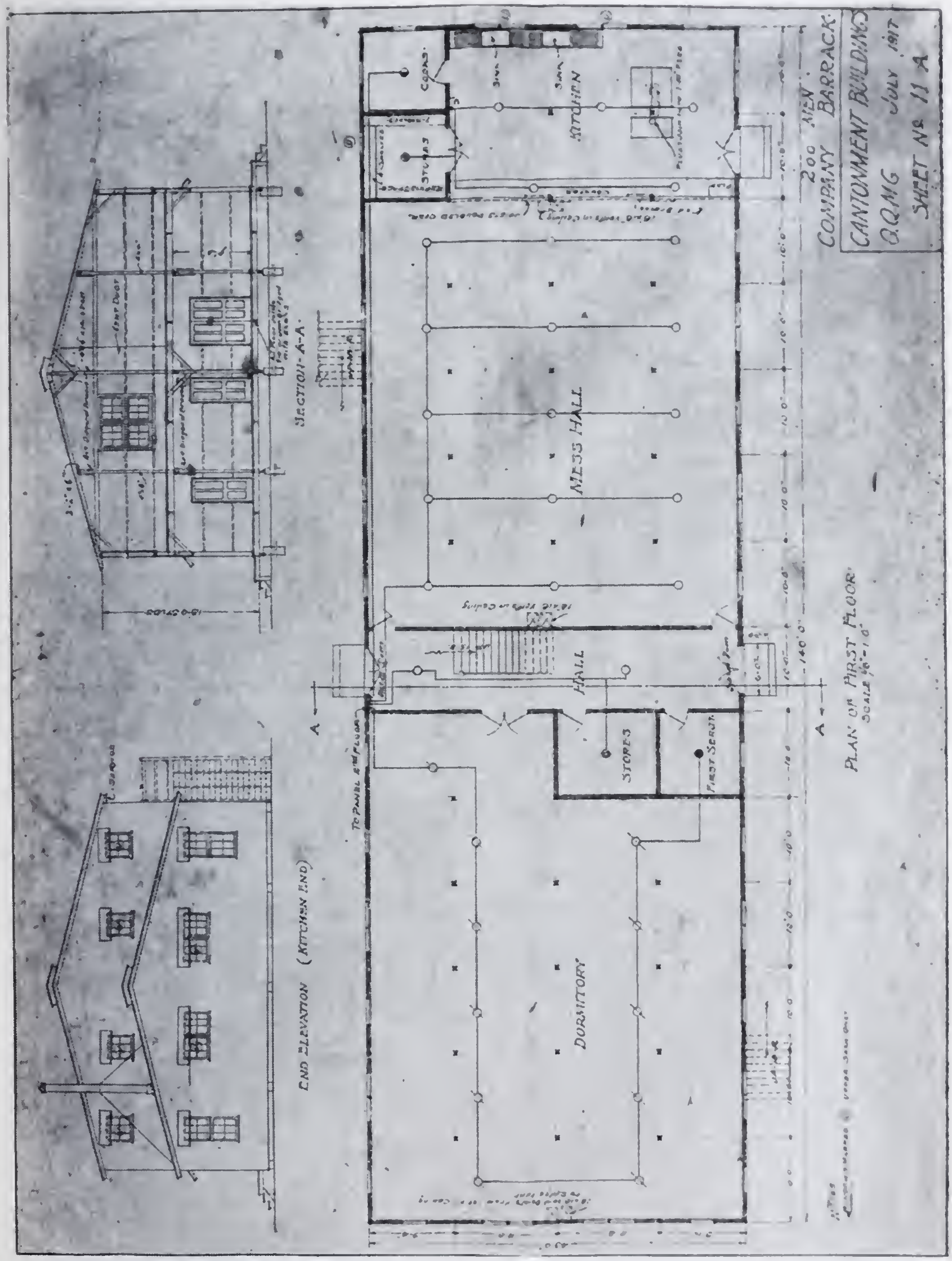

200 MEN
COMPANY BARRACK
CANTONMENT BUILDINGS
O.Q.M.G
JULY 1917
SHEET No 11 A
KITCHEN
COOKS
STORES
MESS HALL
HALL
STORES
FIRST SERGT.
DORMITORY
SECTION-A-A
END ELEVATION (KITCHEN END)
PLAN OF FIRST FLOOR
TO PANEL 2ND FLOOR
A
A

Fig. 4.

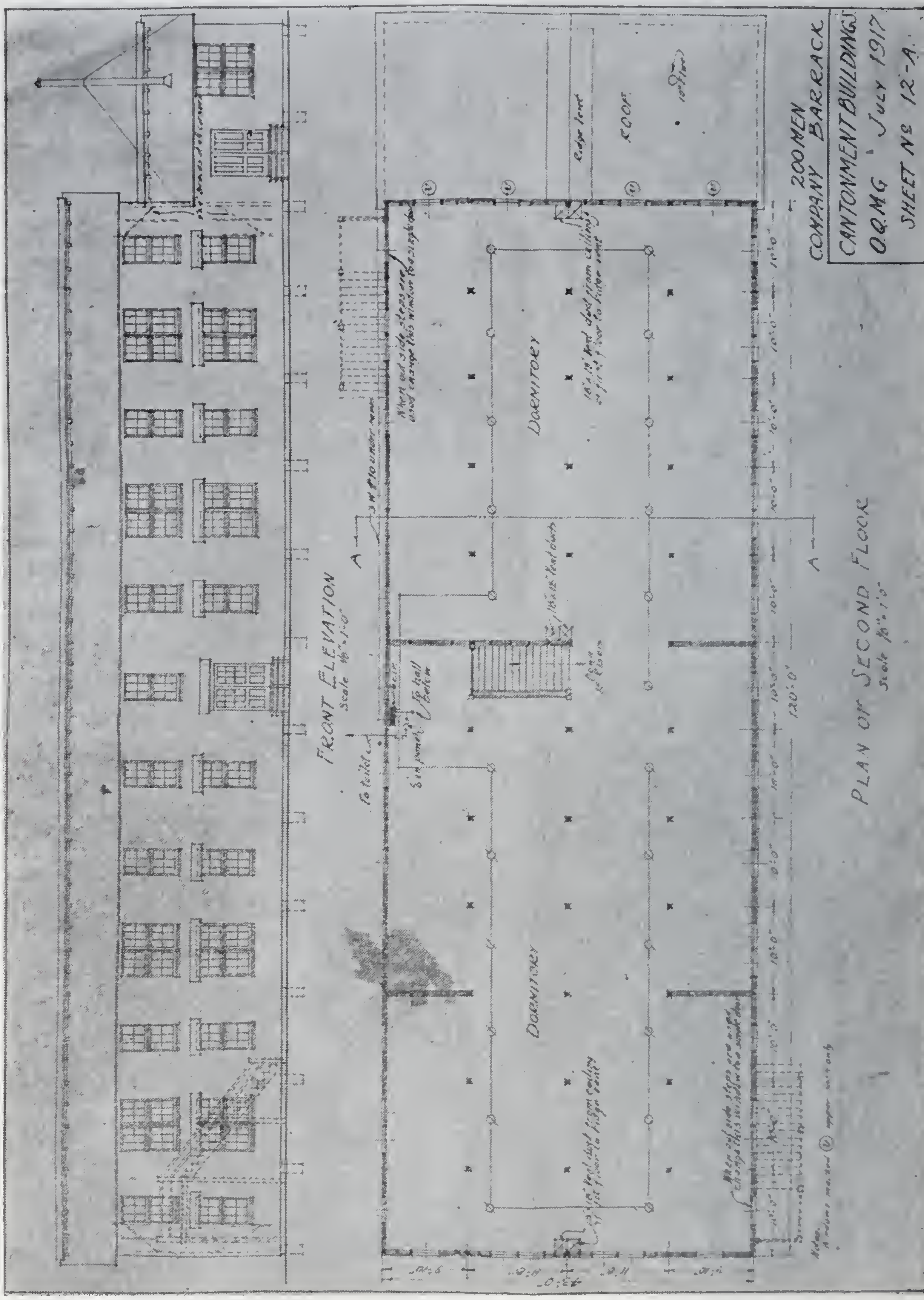

200 MEN
COMPANY BARRACK
CANTONMENT BUILDINGS
O.Q.M.G. JULY 1917
SHEET No 12-A
FRONT ELEVATION
PLAN OF SECOND FLOOR
DORMITORY
DORMITORY
ROOF
120'-0"

Fig. 5.

The newer type of barracks building, arranged in the embarkation camps and in the training battalion units of the cantonments, provides for 66 men in each of the four buildings assigned to the new company. These buildings are 30 by 60 feet, and no kitchen or mess hall is contained or attached. There are provisions for messing the entire company in a separate building.

Officers' quarters are provided in a building 20 feet wide and generally 100 feet long having a kitchen at one end with a mess hall adjoining. Small individual rooms are provided for each officer, so that privacy is obtained, as distinguished from the non-commissioned men. In general the rooms are about eight by ten feet in size.

7. *Electric Light and Power System.* It was necessary at each place to provide electric current for lighting, for pumping water and for other miscellaneous uses. It was also necessary to provide a distribution system, with poles, such that the telephone, telegraph, and fire-alarm systems could be carried to convenient places about the grounds. At Camp Meade 3-phase, 25-cycle, electric current was supplied by the W. B. & A. system, at 33 000 volts. This was stepped down to 2200 volts at a transformer station within the Government reservation and located along the railway right of way, at which point 850-kilovolt-ampere transformers were provided, with suitable appurtenances in the building. At Camp McClellan the current is obtained from the Alabama Power Company, through a 22 000-volt, 3-phase, 60-cycle line, connected to the 110 000-volt transmission line from the hydro-electric plant at Lock no. 12 on the Coosa River.

The system used at both camps is 3-phase, 2200-volt. The problem was to distribute the current so that the various exterior and interior lighting circuits of the regimental units should be separated; with pole transformers at convenient places in the camp. In view of the addition of several units after the original plans were made, it was quite gratifying that for this system, as well as for the water and sewer systems, provision had originally been made sufficient in capacity to take care of the additions, without requiring extreme changes in the main features.

Pumping stations were operated by 440 and 220-volt motors, depending upon that which could be most readily obtained. The power at the low-lift pumping station at Camp Meade is 140 horsepower, and at the high-lift station 370 horsepower. It is also necessary to provide power for the laundry, refrigerating plant, bakery and several other smaller units. At Camp McClellan it was necessary to supply power at the existing waterworks station of the Alabama Water Company; also at a booster station located a short distance from camp. Thorough lighting is provided in all of the buildings of the cantonment; the lights, however, must be shut of at ten o'clock at night. Similarly, lighting is provided in tents as well as buildings at the National Guard camp. In the electrical system at Camp Meade there are 1300 poles, 312 street lights, 143 transformers and 120 miles, or 95 tons, of copper wire.

The heating of barracks in all but the four northern cantonments was accomplished by drum stoves—one stove in either end of each story of the building—four in all. These stoves were of large type, about four feet in diameter and five feet high, capable of being operated with either wood or coal. The officers' quarters were heated by separate steam plants contained in small buildings immediately adjoining. In the four northern cantonments, regimental steam-heating plants were used for heating all the buildings; and in all base hospitals similar plants were used, with pipes conveying steam and hot water to all the buildings of the hospital unit.

8. *Water-Supply System.* The water-supply system at the various camps is one of the most important and troublesome public utilities for which to provide. The requirements are unusual and exacting; even with the assistance of military discipline which can be counted upon to make effective rules and regulations, in regard to conservation and use, which would be considered onerous in civil life. For instance, all fixtures are to be shut off when not in actual use; and details of guards for inspection, and for carrying out this order, are provided. The fire-control regulations provide that, as soon as the fire-alarm is sounded, all fixtures not to be used for the purpose of fire fighting or

cooking shall be kept shut off by guards designated for this purpose. The different brigade units are required to make use of shower-baths at different hours in order to avoid a heavy peak-load. Miscellaneous uses such as sprinkling roads and watering animals shall be so timed as not to coincide with peaks resulting from the heavy draft from showers.

The following criteria were used as a basis of design. The rate of use to be provided for is 55 gallons per capita per day, and provision shall be made, through storage or otherwise, to furnish water to the distribution system, on three minutes' notice and for a period of one hour, at rates 2.85 times as great—that is, at the rate of 5000 gallons per minute for a cantonment of 45 000 men. The design of the distribution system was such as to provide a pressure of not less than 60 pounds nor more than 85 pounds per square inch. The latter criterion was due to the fact that the wood-stave pipe provided was not designed for working pressure above the latter.

Typical water-supply plans, as well as sewer plans, were furnished from Washington. The lay-out was adhered to as closely as possible, the unique features being that—partly on account of the largest size pipe available being 12 inches and partly on account of the desirability of furnishing two main lines of distribution—it was provided that there should be two mains leading from the supply lines, passing through each of the loops of the "U"-shaped cantonment; these uniting at each end. Cross-connection mains were also provided. On account of the danger of blowing out wood pipe, curves or bends were avoided, and standard cast-iron fittings used at sharp turns and at all cross connections, and hydrants were also connected by smaller cast-iron fittings. The wood pipe was entered into the hub of the cast-iron pipe and in all cases a good tight fit was made.

On account of the lack of sufficient farm wells for supply of water during the construction period, and also due to the fact that we were advised by geologists of the United States Geological Survey and of the state of Maryland that there were various zones of water-bearing strata underlying the territory, it was first thought that we could obtain not only the water for construction purposes, but possibly also the permanent water-supply

by drilling wells into these underground sources. Altogether 11 wells were drilled, varying in depth from 50 to 600 feet. The quantities of water thus obtained were small and not up to expectations, varying from 20 to 100 gallons per minute from different wells. Although these wells helped out materially during the construction period, it was soon realized that we should not be able to derive a permanent supply in time, or of sufficient quantity, to supply the entire cantonment. Some of the wells, however, were equipped with gasoline-engine pumps, were used for a period, and are still available as a reserve which can be pumped into the distribution system in cases of emergency and for use up to an amount of about 800 000 gallons per day. All of these wells were located at convenient and strategic points in the reservation, so that in case of failure of the main supply or need of augmentation they will be readily available for assistance.

Permanent supply of water was obtained from the Little Patuxent River by means of an intake located at Welch's Bridge near Portland station on the W. B. & A. Railroad. The stream has a drainage area of 128 square miles at this point, with a population of 53 persons per square mile. Most of the above communities, however, have dry closets and privies, most of the population being of a farming and rural character. There are two institutions, with a population of somewhat less than 900, discharging sewage directly into tributaries of the Little Patuxent River. Sewage disposal plants are in use at both of these institutions. The extreme minimum stream flow of the Little Patuxent River at Welch's Bridge is estimated at 4 000 000 gallons per day, with an ordinary minimum of about 6 000 000 gallons.

Various studies and estimates were made as to the best means of supplying water—in all eight estimates of cost—with the conclusion that the wisest plan would be to adopt a system involving an intake dam, built at the site chosen; with a low-lift pumping station conveying water to a filtration plant, and a high-lift station located immediately west of the lower leg of the cantonment, on the west side near the Portland road, and thence distributing through both sides of the reservation and to storage tanks located upon a hill at the extreme northern end of the property.

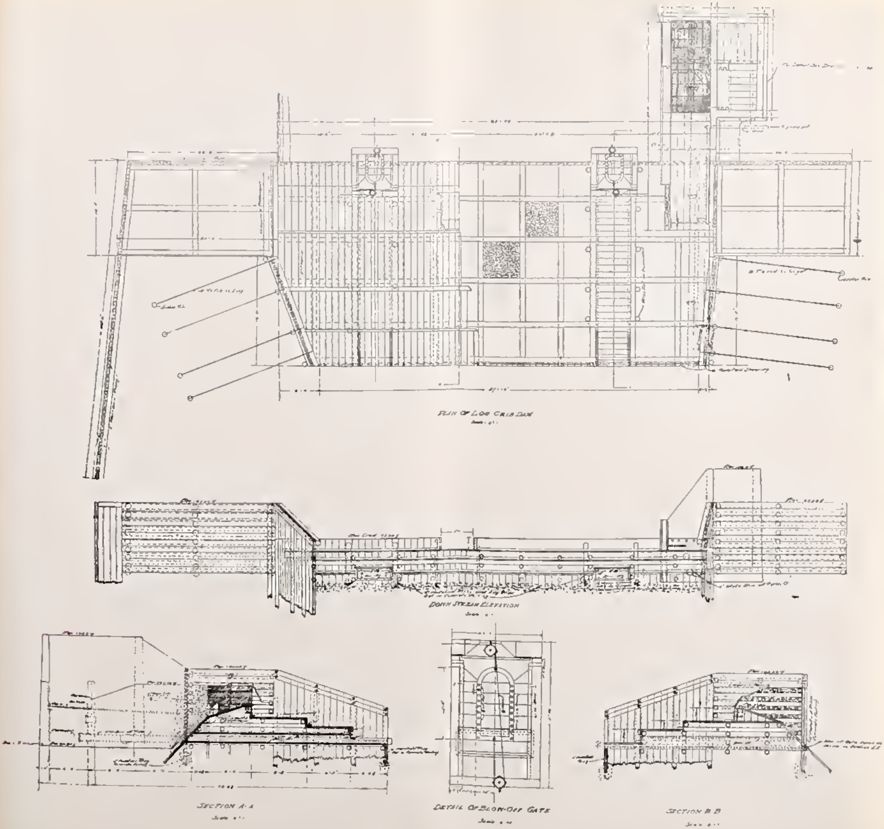

PLAN OF LOG CRIB DAM
DOWN STREAM ELEVATION
SECTION A-A
DETAIL OF BLOW-OFF GATE
SECTION B-B

Fig. 7.

Various studies were made for a dam at Welch's Bridge, resulting in a conclusion on the part of the engineering department that a concrete dam was desirable; but because of the expressed desire of the contractor and his opinion that speed could be obtained and materials more readily secured by building a log-crib, rock-filled dam, this latter type was used. The dam as built was 106 feet long and seven feet high, thoroughly covered with planking, and with three steps on the down-stream side. The intake chamber, which is located adjoining, provides for flowing water over horizontal gratings to the suction well below, from which it is drawn by independent suction pipes to each of the three pumps located in the low-lift station. Views of the complete dam and intake are shown in Fig. 6–7.

Fig. 6.

The work on the dam started August 15 and was completed October 1. The filter plant was ordered July 18, the first shipment was on August 8, and the last car of sand was shipped on August 18. The excavation for the filter plant foundation started August 17, the last order for connecting pipe was placed August 18 and the filter plant was placed in operation September 19. The first troops arrived at one o'clock on that day, and the water became available in the distribution system about seven p. m.

This station provides for three centrifugal pumps—two of 1000, and one of 750 gallons per minute capacity—of the Lawrence Machine Company type, rated for a 100-foot head. All are driven with belt connections from electric motors located on a floor elevated seven feet above the pump pit. From the low-lift station, water passes through two 12-inch, wood-stave pipes about 3400 feet long, to the filter plant, and thence by gravity to the high-lift station.

The filter plant consists of two wooden sedimentation tanks, 20 feet in diameter and 20 feet high—each having a capacity of 100 000 gallons—placed on concrete foundations; and six wooden tub filters, 15 feet in diameter and 7½ feet high, each of 500 000 gallons daily capacity. These filter tubs are located in a building, 48 by 64 feet, with an adjoining alum storage shed, 25 by 30 feet. The filters are the high-velocity wash type, having 15 inches of coarse gravel and 24 inches of sand. They are washed by direct pumping, capable of delivering 2700 gallons per minute. Sterilization equipment is provided by two Electro Bleaching Gas Company's liquid chlorin machines, type F, of the solution feed, which discharge into the effluent main. The clear-water storage

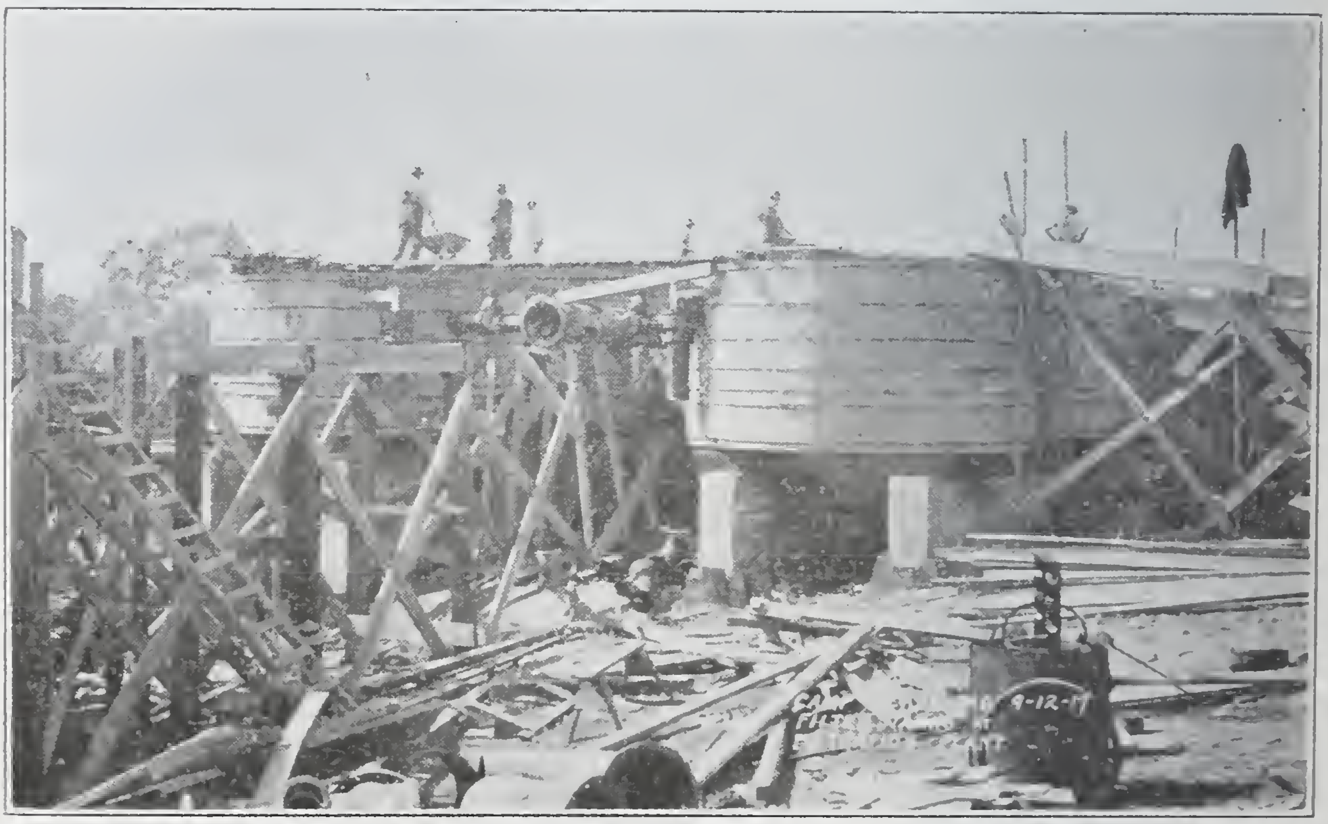


Fig. 8.

is provided for by two 100 000-gallon tanks—similar to the sedimentation tanks—also supported by concrete pedestals. See Fig. 8.

Fig. 9

The high-lift pumping station is located immediately west of the northern clear-water tank, a short distance away. The sloping contour of the hillside was adapted to the convenient and economical placing of all of the tanks and the filter plant so as to avoid excessive grading. This pumping station is a frame structure, 29 by 69 feet, containing one 750-gallon and two 1000-gallon-per-minute Gould two-stage centrifugal pumps, designed for a head of 250 feet; also the two wash-water pumps above mentioned. All are driven by electric motors operated by 440-volt current. The wash-water motors are driven by 2200-volt current. Views of the filter plant and pumping station are shown in Fig. 8–10.

Fig. 10

The discharge from the high-lift pump passes through 8-inch mains into two 14-inch mains, cross connected by a 12-inch supply main. The two 14-inch mains immediately connect with two 12-inch mains within a short distance. The 14-inch header adjacent to the pumping station contains a venturi meter of the "Simplex" type, with a recorder. One of the 12-inch mains runs to the distribution system between the Second and Third Infantry and connects to the 8-inch outside-loop main and

the 10-inch inside-loop main, thus conveying water into the southerly portion of the reservation. The northern 12-inch force main is carried to the Fifth Infantry and connects with the 10-inch outside-loop main and the 10-inch inside-loop main, and thence distributes water to the northerly portion of the cantonment. Various cross connections convey water to the loop mains on the easterly side.

The storage tanks, four in number, each of 100 000 gallons capacity, are located on Hill 300 on the northernmost land of the reservation. These tanks are 36½ feet in diameter and 14 feet deep, placed on trestles 26 feet high (See Fig. 11). The

Fig. 11.

static pressure thus obtained in the lower end of the cantonment, near the divisional storehouse, is somewhat higher than the requirements, being about 90 to 95 pounds.

The total length of distribution pipe is 106 000 feet, or about 20 miles. There are also 95 000 feet of service pipe, 257 hydrants and 177 double gate-valves. As an indication that the system is well built and that wood pipe can be successfully laid without excessive leakage, we are glad to report that the average use per capita in October was 25 gallons, and in November about 30 gallons; the maximum being 42, and the minimum 13½ gallons.

The requirements for use of water at Camp McClellan were somewhat less than for a cantonment; being 30 gallons per capita on an average, and 2.85 times this average as a maximum for one hour. The system at Camp McClellan was therefore designed for a daily delivery of 1 500 000 gallons per day instead of 2 250 000 gallons used at Camp Meade.

At the southern camp there were numerous spring supplies which probably would be able to yield 2 000 000 gallons a day, but they were situated far apart and would require pumping; and it was apparent that they could not be made available, with the necessary construction works, within the time limit set. The city of Anniston is supplied by the Alabama Water Company, deriving water from Coldwater Spring, which has a minimum flow, out of the lime channel rocks of the mountain south of Anniston, of 25 000 000 gallons per day. This water, after passing through the distribution system of the city, arrives at a place called Blue Mountain, whence it would be conveyed to the camp along the right of way of the Southern Railway, a distance of 23 800 feet. A 10-inch line was therefore laid from this place to the reservation. There was also installed a necessary, additional, motor-driven, centrifugal pump, placed at the Coldwater Station of the water company, and a booster station of 2000 gallons per minute capacity was built near the Blue Mountain connection.

This latter pumping station lifts the water against a 225-foot head and, after passing through the camp reservation, the surplus water is stored in an 800 000-gallon reservoir, located on a hill north of the camp site. It is thus seen that the details of the system are not as comprehensive as those for Camp Meade. The reason for the lesser use of water is that it is supplied only to the kitchens of the mess halls, and the showers, these units being located at opposite ends of the company streets. Fire connections were also placed at essential points over the reservation.

9. *Sewerage System and Sewage-Treatment Plant.* Cantonments require complete sewerage systems with satisfactory disposal so as not to cause nuisances; therefore treatment is required in varying degrees. National Guard camps, because of having latrines, do not have sewer systems; except that base hos-

pitals have since been provided with the necessary plumbing, and therefore sewer systems and sewage-treatment plants are being installed.

The general instructions for the sewerage system provided that no surface drainage shall be admitted, and that all building connections shall be six-inch, except in a few cases where four-inch may be used for kitchens. Cover for pipe lines shall be two feet. Typical plans showed tentatively the arrangement of house connections and laterals; but this, of course, was departed from somewhat when the group plan was determined. The allowances for sewage were 1½ second-feet for one regiment, 4½ second-feet for three regiments, reaching 8 second-feet for 10 regiments. The minimum grades permissible were:

0.5 for 6-inch pipe
0.4 for 8-inch pipe
0.3 for 10-inch pipe
0.2 for 12-inch pipe
0.12 for 24-inch pipe

First-class, salt-glazed, vitrified, clay pipe of standard design was used. In order to avoid inverted siphons the terra-cotta pipe was in two places laid across swamps upon wooden trestles.

It was early determined that the best design of sewer would provide for two main trunks—one extending down each side of the Rogue's Harbor branch—with the laterals from the various regiments running into these trunks at convenient places. In order to maintain uniform grades it was necessary in many cases to detour these main trunks so as to follow the contours, and it was possible to complete the system without excessive cutting in many places. After the main trunk sewers were designed and the laying almost completed, it was decided to add to the cantonment the base-hospital unit and six training battalions. Because of the liberality of our design it was possible to take care of the sewage from these units with but few changes—these being in the sizes of some of the upper lines.

Sewer trenches as well as water trenches were dug by machines of the caterpillar type. There were six in use at one

time; the average amount of sewer-pipe used was 1770 feet per day, and a miximum of 3300 feet was laid in one day. The total amount of sewer-pipe used was 31 miles, or 2000 tons.

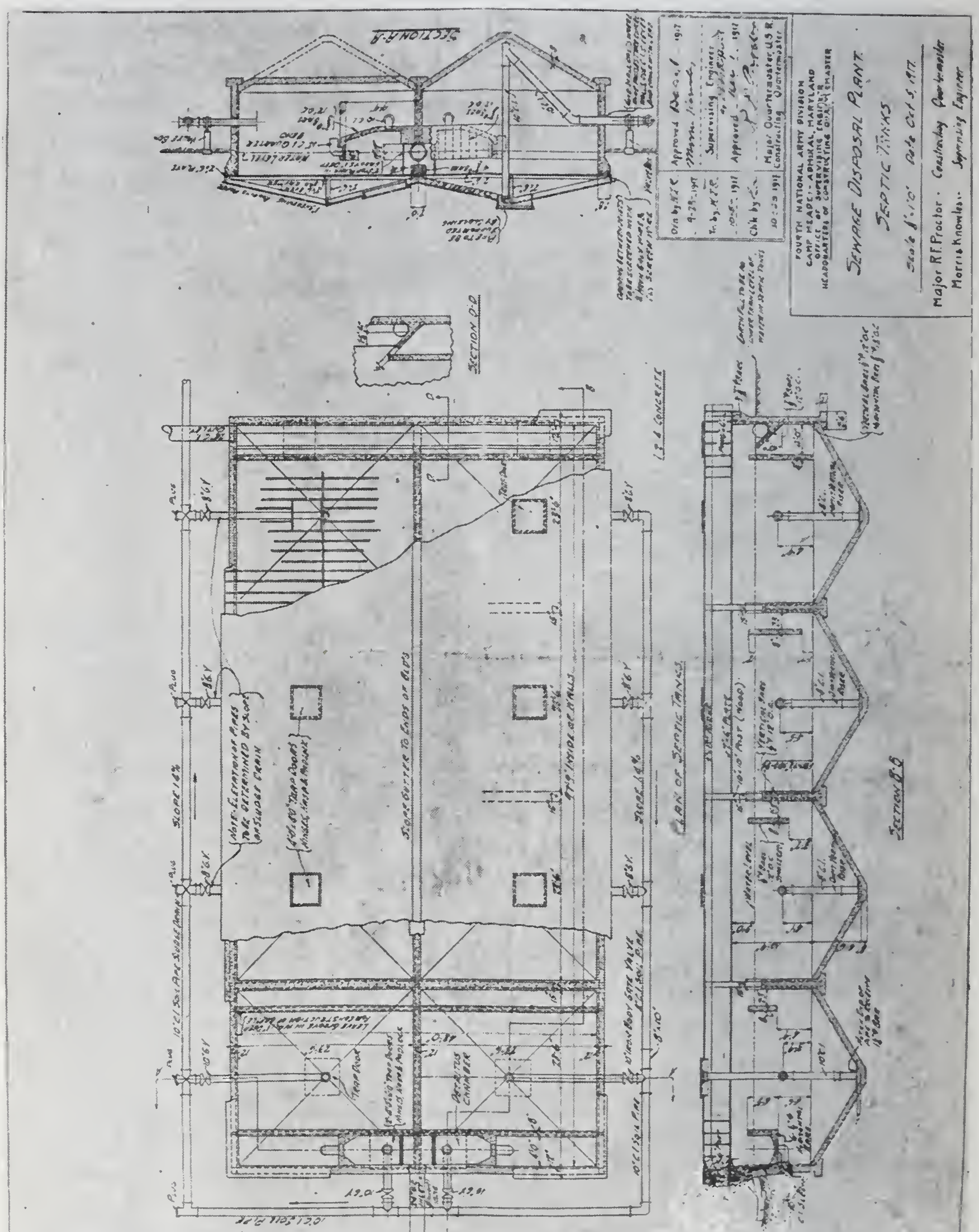


Fig. 12.

The sewage-treatment plant, as originally designed on the plans sent from Washington, provided for single-story tanks with pyramidal bottoms. The gross tank capacity provided was at the rate of 10 gallons per soldier. Trickling filters were provided in some instances but it was decided at Camp Meade, after consultation with the authorities of the Maryland State Board of Health, that the effluent from the tanks, when sterilized by the use of chlorin gas could be admitted to the Little Patuxent River about 10 000 feet away. It was therefore decided to omit the filters and convey the effluent, through a 24-inch terra-cotta pipe outfall sewer, to the river—instead of disposing in the Rogue's Harbor branch near at hand.

The sewage-treatment plant was located approximately 3500 feet south of Admiral station and within the maneuvering ground. The tanks are built of reinforced concrete, 23½ feet square and 10 feet 9 inches deep, to the top of the bottom slope, with the center 7 feet lower. Eight tanks only were built; it is provided, however, that four more may be built adjoining, and these, as well as the filters, will probably be required in the future. The arrangement of the tanks is shown in Fig. 12.

Additional disposal and treatment units were planned for three buildings in the bakery group—so situated that sewage could not readily enter the system—and at the water filtration plant and the horse remount depot, which were located some distance from the cantonment system.

ELECTRICITY AS A SUBSTITUTE FOR NATURAL GAS FOR HEATING PURPOSES

By Frank Thornton, Jr.*

The substitution of electricity as a heating agent instead of natural gas—or of any form of fuel, for that matter—is a much more complicated problem than the substitution of one fuel for another. In the case of fluid fuels—natural gas, artificial gas, crude oil, refined oil, powdered coal, etc.—the heating methods are more or less identical in principle. The heat is generated by the combustion of the fuel. This fuel is fed into the furnace through suitable pipes and nozzles after being properly mixed with air. The method of doing this, and the design of the apparatus, may vary according to the nature of the fuel, but there need usually be little or no modification of the furnace proper, to accommodate any of the common forms of fluid fuels. Even solid fuels may be grouped with the fluid fuels to some extent in this respect because often the fire acts simply as a gas generator and the flaming gases flow into the heating chamber to accomplish the desired function of heating.

The generation of heat by the combustion of fuels takes place at high temperatures, the flame temperature being partially controllable but having a maximum and a minimum limit. Electrically generated heat may be produced at any desired temperature up to the maximum obtainable from the arc. The method employed for generating the heat, the means of applying it to the desired useful purpose, and the amount of power used, all influence the temperature obtainable. Different applications require the use of entirely different methods. The result of this fundamental difference between heat obtained from the combustion of fuel and that obtained from electricity, is that it very often becomes necessary to develop new designs of apparatus and even

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to modify processes in order to successfully and commercially use electric heat.

Before entering into a discussion of apparatus design it would be well to consider some of the fundamental features of the problem. Electricity for commercial power purposes is obtained from generating plants driven by steam engines, internal combustion engines, or water-power. There are other sources such as wind, wave, tidal, solar, volcanic, and chemical, but these are not to be considered as important sources at present. Steam may be generated by burning any of the fuels, such as coal, gas, or oil.

The first question which arises in connection with the consideration of any substitution problem is naturally the one of cost. We know gas costs a certain amount per thousand cubic feet at a certain heat content per cubic foot. Electricity costs a certain amount per kilowatt-hour at 3412 B.t.u. each. It is easy to figure out what a B.t.u. of heat energy costs in each case. Natural gas of 1000 B.t.u. per cubic foot at 27 cents per thousand feet means 27 cents per million B.t.u. Electricity at one cent per kilowatt-hour means $2.94 per million B.t.u. Thus, electrically generated heat units are nearly eleven times as expensive as those produced by the combustion of gas.

Let us take the case of a coal-fired, steam-engine-driven, electric generating plant. The boiler efficiency will be in the neighborhood of 65 per cent. and the engine efficiency 15 per cent.; generation and transmission efficiency 85 per cent.; total over all efficiency $0.65 \times 0.15 \times 0.85 = 0.083$ (approximate average value). If the coal has a thermal value of 14 000 B.t.u. per pound, this means that $14\,000 \times 0.083$, or 1162 B.t.u., are obtained for each pound of coal burned. Similar reasoning can be applied to any kind of fuel-fired, electricity-generating plant, with similar results. By such calculations it appears to be entirely beyond question that the utilization of the heat of combustion, directly as the heating agent, is more economical than to use electricity generated by this method.

In the preceding statements, no mention has been made of the efficiency of the application of heat. Here, however, we are not

always on known ground. The thermal efficiency of steam generating plants has been the subject of careful investigation and the design of such plants has been given so much attention that we can make fairly well justified assumptions concerning them. In the application of fuel directly to heating purposes, we have some information as to requirements and thermal efficiencies although varying conditions introduce wide variations in results. Furthermore, thermal efficiency is measured by results obtained and cannot, in general, be considered separately from results. In connection with the application of electricity to heating purposes, we are still unfamiliar with thermal efficiency and with its influence on results.

It is necessary, therefore, in considering the application of electric heat, to analyze each problem completely. If it is desired to heat an object, the fuel is usually burned in a chamber which is sufficiently large to readily admit the object. The maximum temperature to which that body can be heated is limited by the temperature of the flue-gases passing out of the chamber. If only a low temperature is desired in the body, a large proportion of the heat may be removed from the flue-gases and they may be allowed to pass out fairly cool. If a high temperature is desired in the body, then the flue-gases must go out at a high temperature and therefore carry away most of the heat of combustion with them. Thus, in the case of a steam-boiler—a low-temperature application—it may be possible to obtain in the steam as much as 80 per cent. of the heat of combustion from the fuel; but in a high-temperature forge furnace only two or three per cent. of the heat of combustion may be obtained in the heated steel as taken from the furnace.

An electric heating equipment is not subject to this flue-gas loss and may therefore be operated at a higher thermal efficiency. An electrically heated steam generator can be made to give efficiencies as high as 95 per cent., while a high-temperature forge furnace may operate at 50 to 60 per cent. Even though the efficiency of application was high in the case of electricity, the cost of a heat unit has been so much higher than that obtained from gas, that it has long been considered uneconomical from a cost standpoint except in very special cases. It has not been many

years since costs of natural gas in the neighborhood of five cents per thousand cubic feet, and of electricity in the neighborhood of five cents per kilowatt-hour were common. Under these conditions it is obvious that on a strictly thermal consideration electricity could scarcely be used except when there was some very unusual condition.

All manufacturing operations are made up of several different elements such as material, equipment, labor, motive power, heat, time, etc. A change in any one of these elements may influence the first result, not only directly but also indirectly, by changing one or more of the other elements. It is good business to use expensive equipment, if thereby the total cost of the product can be materially lessened by increasing production. Examples of reductions in costs made in this and similar ways are so numerous that they need not be recounted here. It is only necessary to recognize the fact that there are such possibilities.

Unusual conditions, bringing about such results, have already developed in connection with so many commercial applications of electricity to heating that it may be well to consider a few of them at this time. They are so varied in nature as to be almost impossible of classification.

An application which has developed very rapidly in the last few years is arc welding. Welding two pieces of metal together with an ordinary flame is accomplished by heating almost to the melting temperature the portions to be welded, joining them together, and hammering them into a homogeneous joint as they cool. This requires a great deal of time and heat and is extremely inefficient thermally. By means of the electric arc the operation is quickly and accurately performed and only the metal adjacent to the surfaces to be welded, is heated. The saving in time, labor, and heat energy is enormous. Furthermore, there are many operations both of a repair nature and of a constructive nature which can be performed only with the electric arc. It is undoubtedly much better, from the point of view of fuel economy as well as financial economy, to repair an expensive broken casting, for instance, by the expenditure of a little time and electrical energy, than to scrap it and spend a large amount of heat and labor to remelt and recast it.

Another application of electric heating which has recently grown rapidly is that of japanning, enameling, and other forms of industrial baking processes. Here, numerous conditions have been determining factors. Gas ovens of the kiln type were the rule. These were charged with the material on racks or carried in by hand. Control of temperatures and regulation of the atmosphere in the ovens were uncertain. Indirect heating was inefficient and direct heating introduced the products of combustion into the chamber containing the work and created undue air circulation, carrying dust particles on to the work. Usually the japan or enamel contained very combustible and volatile matter and the fire hazard was great. The presence of the gas itself introduced fire and explosion risks. The introduction of electric heating made possible more accurate control of temperatures, better distribution of the heat, greater safety, better regulation of the atmosphere and also the development of continuous-process ovens. The result has been a better product, lower cost of production, and greater efficiency of heat application. This has come about, even though, from purely thermal considerations on the basis of operations as they existed when using gas, it appeared that electricity would be from two to four times as expensive.

Industrial baking applications have been reported so frequently in the technical press that it is unnecessary to list them here. They extend from the baking of enamel on automobile bodies to the japanning of baby carriages, and from the setting of paint on washing machines to the baking of doll heads.

Similar conditions were found when the application of electricity to hot molding presses was investigated. For hot molding at temperatures above that obtainable from 80-pound steam it had been customary to use gas. Combustion of the gas took place in holes in the press heads, and regulation of the temperature was extremely unsatisfactory. Electric heating was tried and it was found that the uniformity of temperature regulation made possible a large saving in defective product which had formerly been caused by under- and over-baking. The value of the product thus saved was several times the cost of the electric energy consumed. It would have been unprofitable to have used gas even if it had cost nothing.

There are numerous heating applications—and the number is increasing rapidly — which are being more economically, more safely, and more satisfactorily solved by the use of electric heat. In no case have there been wholesale substitutions of electricity for gas throughout any one industry. The problems of proper design of the electric heating equipment, coupled with those of application, have made it necessary to go slowly. Often a process must be handled in an entirely different manner to successfully use electric heat. Naturally those problems have been solved which offered the quickest financial returns to those interested in their solution. Other problems will follow the earlier developments as experience is gained. Changing conditions may also expedite development.

It is safe to say it is now possible to satisfactorily design electric heating apparatus for any application requiring temperatures up to 500 degrees C. (932 degrees F.) and that a larger number of installations is being operated commercially; that many operations requiring temperatures between 500 degrees C. (932 degrees F.) and 900 degrees C. (1650 degrees F.) are being successfully performed; and that operations above 900 degrees C. (1650 degrees F.) and below 1230 degrees C. (2250 degrees F.) seem to be within the range of early possibility.

In the field of high-temperature electrometallurgical operations, such as refining of steel in induction or arc furnaces, much experience has already been gained and there is available fairly good information as to power requirements and the value of the results. The increased value of the product justifies the expenditure required for new equipment as well as the cost of the power and other operating costs. There are many other processes to which electric heat is being applied for similar reasons.

This discussion has been devoted entirely to industrial applications in daily use, most of which are continuous during the period of work, and many of which are continuous throughout the twenty-four hours. There is a big application of heating which has not been mentioned, namely, heating of buildings. This has a seasonal requirement, having the period of maximum demand usually short and having a long period in the summer when there is no demand. It can be supplied economically by electricity only

where the generating stations are without other load in the winter time. This occurs very rarely, but there are a few cases, such as occur in irrigated regions in which water is pumped by electricity which in turn is generated from water-power. Here, it is necessary to keep a load on the plant to prevent freezing in the winter and this can best be accomplished by connecting a heating load to the lines. If a fuel-fired generating station were to be large enough to care for this maximum peak-load which would occur in the coldest winter weather and which would be superimposed on the usual power and lighting load, the investment for the plant and the distribution system would be very high and it is doubtful whether it would be possible to furnish the energy for heating as cheaply as the coal could be bought and burned. Continuity of domestic heating in cold weather is also a vital point. Failure of the power system would bring wholesale disaster to the community relying on it for heat. Not that there are no cases in which such heating can be done satisfactorily, for there are many special cases in which such electric heating of spaces in buildings may be commercially economical. In general, however, at the present time, with Nature's stores of fuel to draw upon, the general substitution of electricity for fuel for this purpose does not seem probable.

Although electric heating has been dubbed "One of the babies in the family of the electrical industries," yet it is an exceedingly healthy and rapidly growing infant. It has been stated recently in the *Electrical World* that there is no single generating station in the country that could supply power simultaneously for all the electric heating apparatus manufactured during the past year.

DISCUSSION

Mr. A. Stucki:* We realize that no claim has been made by the author for greater or even equal efficiency in fuel. None the less, it would be interesting to know just what takes place, for instance, in a heating furnace. If fuel is burned directly, all the heat produced stays in the furnace and practically none is lost. In case radiators are used, the transmission of heat diminishes as the furnace gets hotter. However, none goes to waste, as the heat not given off to the furnace simply stays in the circuit. Now, coming back to the electric current, I should like to know whether all the heat contained in the current is given off until the temperature of the furnace reaches a maximum—at which time it is no doubt shut off—or whether, as the temperature rises, only part of the heat is given off. In this latter case, does part of the current return or is it lost?

Mr. Frank Thornton, Jr.: I am not sure that I understand Mr. Stucki's question. With a steam radiator such as we have in buildings, there is a certain amount of heat in the steam which it receives. The steam condenses and the water goes back to the boiler carrying with it part of the heat. In that case the steam radiator of course gives off heat equal to the amount of heat given up by the steam in condensing, and that represents radiant loss. It also represents the amount of heat which must be supplied by the furnace to the boiler to bring the steam to the proper condition of temperature and pressure to be delivered again to the radiator. Thus the water serves as a transporting medium for the heat, and is in that way analogous to the flow of the electric current. The passage of the electric current through the resistance of the electric radiator gives off heat. As a matter of fact all the electrical energy that goes into the radiator is given off as heat so that the thermal efficiency would be 100 per cent. That is, the amount of heat given off equals the amount of energy put in. Does that answer your question?

*Consulting Engineer, Pittsburgh.

MR. A. STUCKI: As the temperature rises you take up less heat in the electric apparatus. In other words you use less current for the same heat.

MR. FRANK THORNTON, JR: That is a point I did not mention in the paper as I did not intend to go into details.

In any operation which requires the apparatus to be heated up from the cold, there is naturally considerable absorption in the apparatus itself, and that heat can not usually be recovered for useful work. If the electric energy goes in at a constant rate—and it will go in at a constant rate when both the resistance and the voltage are constant—it means a continuous increase of temperature until radiation losses equal input. The temperature is not a constant function as it would be in a constant-pressure steam-heating system. There is a fundamental difference between the two systems.

MR. H. D. JAMES:* I believe Mr. Stucki wished to know how we regulate the heat in the oven and whether the electric power is wasted if there is surplus capacity. Usually there is a surplus capacity in the heating element and the temperature of the oven is regulated either by interrupting the current or by reducing the amount of current. In no case is power wasted. You have, due to the transmission of the power, a constant loss which corresponds to the loss in the steam pipes leading to a radiator. The loss, however, is much less than with steam heat.

MR. W. E. SNYDER:† I would like to know more about the details of voltage and quantity used in various kinds of heaters. I do not know whether it is possible to get definite information of that kind without going too much into the description of the apparatus, but what I should like to know is exactly how to calculate this. For instance we want to put in a small heater in a garage or a small japanning furnace. I should like to know what to do to find out what the cost will be for the current per hour for the time the apparatus is in use; what kind of current is

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†Mechanical Engineer, American Steel & Wire Co., Pittsburgh.

required; what is the character of the load; the various steps of investigation to learn how much load to put in per furnace; the character of the current; the voltage; and how much will be consumed, etc.

Mr. Frank Thornton, Jr.: That is pretty hard to answer without details to work on. It might be of interest to take the blackboard and arbitrarily assume some figures. Almost any assumption will do for the purpose. We might assume that it is a japanning oven, six by six by six feet. Assume that the racks weigh 200 pounds and that the charge is 600 pounds of steel punchings; that the maximum temperature is 1400 degrees F. and the initial temperature 60 degrees F. Then assume that the time period is 30 minutes. Varying conditions often determine these points. Under these conditions a japanning operation consists practically in bringing the temperature up only to the maximum temperature and then pulling the work out and putting in a new charge. That would give a simple problem. First, we must supply the energy lost by radiation. Assume that the outfit is operating, and do not take into consideration the initial heating up period. If it is going to operate ten hours a day it does not make much difference whether it takes half or three-quarters of an hour to bring it up to temperature. Second, we must supply ventilation. Third, we must supply absorption, both in the charge or work, and in the racks. The first item is dependent upon the construction of the oven. If the oven has just an iron frame and sheet-iron sides with no insulation, it will have a big radiation loss. With improved insulation it will cut down the loss in proportion. We will assume that it will be insulated and that the radiation loss might amount to 30 watts per hour per square foot. At this rate 216 square feet would radiate about six kilowatts, or about eight horsepower. The ventilation will, of course, also depend on the quality of the work, how dry it is when put in, and various other factors. It might amount to six changes of air per half hour. We will assume that it is four kilowatts.

Regarding absorption in the load. We have 600 pounds of material and 200 pounds of racks—or 800 pounds of iron to be heated from 60 to 400 degrees F. every half hour. That means

a temperature rise of 340 degrees F. The specific heat is approximately 0.125, or 1/8, and 340 × 800 × 1/8 equals 34 000 B.t.u., or about 10 kilowatt-hours. This is for a half hour, which gives a total of 20 kilowatts of electricity input. The total input for absorption, ventilation, and radiation is thus about 30 kilowatts. Such electric heating apparatus is usually operated on 440 volts or less.

That is roughly the process we would go through, and the same principle could be applied to any heat problem. After the input has been determined, the number of heating units capacity sufficient to furnish that total number of kilowatts can easily be determined.

The other part of the question, as to the uniformity of the power and the character of control, would depend on the character of this baking load. If it were a simple baking operation—bringing the material up to baking temperature and taking it out—it would be controlled by a thermostat set at maximum temperature to throw the current off; or it might turn the power on and off to maintain the temperature constant for a certain length of time if that were necessary. The form of the control would depend on the condition of the process and the rapidity with which the charge is to be reloaded, and those factors would in turn determine the total amount of power.

Mr. Theodore J. Vollkommer:* I am not familiar with baking furnaces but am to some extent familiar with furnaces of much higher temperature—900 to 1100 degrees C. For these temperatures, as far as I know, there have never been any electric muffle furnaces used on a large scale, but it seems to me it would be a very good thing aside from the question of cheapness. But there is one thing about the fuel-fired furnaces in actual use. No matter how carefully they are designed there is generally near the door an underheated zone which is generally about 10 feet long, and in the rear part of the furnace an overheated zone. In such cases it seems to me that, though the cost of electric heating is greater than gas or coal heating, possibly the advantage of having a uniform heat throughout the furnace might compensate

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for this drawback. But I do not know that there is any large electric furnace in use. There are many laboratory furnaces, of course, but nothing on a large scale so far as I know.

Muffle furnaces are generally about 10 feet long and frequently unequally heated, the part near the door being liable to have too low, and the rear part too high a temperature. Also the presence of fuel gases in the muffle (through cracks) is liable to injure the gloss and quality of the enamel ware. Possibly the advantage of obtaining first-class ware, without seconds, might compensate for the higher operating cost of electric muffle furnaces.

Mr. L. C. Frohrieb:* Possibly Mr. Thornton can give us an idea of the temperature to which a large furnace can be satisfactorily heated by the electric method.

Mr. Frank Thornton, Jr.: The statement just made practically covers the ground. For other than laboratory use large furnaces have not really been developed to a commercial point at the present time. But it is also true that electric furnace people recognize the very great opportunity which has just been mentioned, and the possibilities within the range of future development.

Mr. W. E. Snyder: There is one point which frequently arises in comparing various methods, and which came up recently in connection with an electrically heated japanning furnace. To compare the cost of electric heating with that for an oil-fired furnace—using a definite quantity of oil at a certain rate per gallon—it is customary, as has been shown here, to calculate the required electric power, figuring it perhaps at 1 or 1¼ cents per kilowatt-hour, arriving at a result perhaps twice as high as the cost of oil. From this it is argued that the cost of electric heating is disproportionately high. Suppose we consider the expenditure which is to be charged up for electric power. Part of this expenditure is for steam—the cost of steam including boiler house expense, station cost, labor, supplies, etc. None of the various

*Secretary, Federal Engineering Co., Pittsburgh.

items, except fuel, will be materially increased by the addition of an electric furnace, but the sum total of the additional expense at the plant, due to the small added load on the power station, will not equal half the cost of producing the electric power; so that the actual increase in expenditure that is made necessary because of this, rather than the fictitious cost shown on the cost sheet, would bring you to an entirely different conclusion as to the relative cost of heating. That is a condition which I have seen come up time and again, and I should like to have the opinion of some one who has gone into that matter, as to what is the proper course to follow.

MR. H. D. JAMES: If the cost of electric power for a given heating effect is more than for gas heating, the use of electric power should be justified by either an increase in output or an improvement in the quality of the finished article, which would give it an added commercial advantage. Where a certain percentage of the product is lost, due to improper heating, the electric heat will materially reduce this loss and the saving thus effected may justify the difference in cost of power.

Electric heating has been retarded recently, due to the difficulty of obtaining electric power. This is a very unusual condition and is only temporary.

MR. L. C. FROHRIEB: Mr. Snyder's question resolves itself into this. Is it proper to neglect overhead charges in the production of electric current where it is used for a small part of the entire plant, or should the overhead be charged to the rest of the plant?

MR. W. E. SNYDER: Overhead is generally used referring to general expense outside of the cost of product. My point referred to the fact that a part, but by no means the larger part, of the total cost of producing electric power is increased in proportion to the quantity produced. Therefore a misleading conclusion is reached by taking the actual cost ot producing power at the plant when a small addition is put at the furnace, because

hardly any operating expense is increased except the coal item and possibly a very small increase in the repairs or supplies. It is a problem to tell just where to stop if we look at the matter in this way. It is apparent that if we add a large consumer we increase the expense more in proportion than if we add a small one.

MR. H. O. SWOBODA:* Mr. Snyder's statement—that the average cost of electrical energy as calculated in mills from the books of account, might not represent the figure which should properly be applied as the actual cost—is correct to a certain point.

The load of any electric power-plant varies in accordance with the demand made upon it by the various motors, lights and other appliances which are connected to the system, and only in exceptional cases is this load constant. If an electric furnace or any other appliance is added to a plant which is already in existence, and this furnace is operated at a time during which the capacity of the power-plant is not taxed to its maximum by the motors, lights and other appliances, then the actual cost of the electrical energy delivered to the furnace is made up of nothing but the additional consumption of fuel, plus an insignificant percentage for wear and tear of the equipment. Wages, salaries, interest on the investment and depreciation of the equipment need not enter in such a case into the cost of the electrical energy for the furnace, as these items were incurred and *fully* charged as a part of the cost of that energy which is consumed by the other equipment. This method of determining the cost might, of course, be called a discrimination in favor of the furnace at the expense of the other equipment, but the fact nevertheless remains that the costs established in this manner are mathematically correct. The freight rates of railroads and the rates for electrical energy charged by the central stations are built up on the very same principle, and should the railroads and central stations attempt to apply the *same* rate for *all different* customers and consumers, they could not exist—a fact which is well established.

*Consulting Engineer, Empire Building, Pittsburgh.

Of course, should the addition of an electric furnace necessitate an increase in the capacity of the power-plant, then this increase must be taken care of in the cost of the energy for the furnace, although the said increase need not necessarily be in a direct ratio to the increase in the investment.

Central stations recognized these facts long ago and express this recognition very well in their system of charges, called the "demand" system. In accordance with this system one charge is made for the "demand" made by the consumer on the capacity of the central-station plant, and another charge is made in accordance with the number of kilowatt-hours consumed. The combined charges represent the cost of the energy to the consumer. In accordance with this system, a *large* demand on the capacity of the plant with a *short* period of energy consumption results in a comparatively high energy bill; whereas for a *small* demand with a *long* period of energy consumption the bill is low. This method of charging has been adopted and is fully justified, as it increases the "load-factor" of the central station and reduces the number of idle hours for the power equipment.

MR. FRANK THORNTON, JR.: The point about the central station having various rates for various kinds of service, brings out the fact that the central station had at one time to face that very problem, and to get away from a fixed standard of price for electric energy, by adopting a graduated scale depending on various conditions, such as the time at which the peak-load came; the character of the load, and other conditions. While that is now being done quite generally in central-station supply business I do not believe there are very many factories that make use of the graduated charge, depending on the character of the load, in different applications or in different departments. It is done in some cases in large factories where they have a great variety of supply, and the expense of serving different departments may be different. This is something that might well be considered in connection with the furnishing of power by generating plants in manufacturing establishments, to various departments, in order to equalize the cost accounting which has just been mentioned.

Mr. H. P. Smith:* Last winter when the gas shortage seemed inevitable, an investigation for a substitute for our natural-gas-fired, rivet-heating furnaces was made and included a test of an electric furnace. The cost of heating with electricity compared favorably with our present practice. This did not take into consideration any improvement in the product which was a decided feature of many of the furnaces mentioned this evening. The thermal efficiency, when heating rivets, is about 50 per cent. This is low on account of the door of the furnace being opened frequently. When used for heat-treating steel and melting non-ferrous metals, this furnace reaches 90 per cent. thermal efficiency. Temperatures above 2500 degrees F. are readily obtained. The electric furnace in which the rivets were heated is six feet long, 30 inches wide and 18 inches high, and has three doors at the front. Seventy rivets, weighing 45 pounds, were heated to the proper temperature for driving, in each compartment, every 15 minutes. The furnace required 40 kilowatts.

Mr. L. C. Frohrieb: In this riveting furnace is there any danger of burning the rivets, or is the heat maintained at the predetermined point no matter how long the rivets are left in the furnace?

Mr. H. P. Smith: The electric furnaces are equipped with auto-transformers, having the high-tension side wound for 440 volts and the low-tension side wound for the voltage to suit the furnace, which in this case was about 60 volts. The furnaces have no vents and, where the operation permits, the doors are kept closed and there is no change of air with an accompaying loss of power. Besides operating on as low cost as with fuel-oil, an additional saving is made when the furnaces are used for the purpose for which they are particularly adapted; that is, heat-treating steel and melting copper and precious metals.

Mr. L. C. Frohrieb: On what basis of current price was that?

Mr. H. P. Smith: Our estimates were based on energy at

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one-half cent per kilowatt-hour. The electric-furnace load-factor is unusually high, and the power factor 100 per cent. At the time the electric furnaces were under consideration, a low rate for energy could be obtained for the above reasons.

The heating element in this furnace was a semi-circular trough of four inches radius and about four feet long, having an electric terminal at either end, and filled with graphite. The current from the auto-transformers passes through the graphite, causing it to become incandescent. The loss of graphite, due to oxidation in furnaces used for rivet heating, was estimated to cost 30 cents per ton of metal heated. When operating with the door closed, the loss of graphite by oxidation would be very much less. The furnace has an arched roof and the heating is indirect. The walls are thoroughly insulated and after the furnace had been in operation six hours, the walls of the furnace were but slightly above the room temperature.

Mr. H. O. Swoboda: As additional explanation to the statements previously made by me about the charges for electrical energy made by central stations, attention is called to two different schedules of the Duquesne Light Company of this city, both based on the principle of the "demand" system. Schedule F—"wholesale light and power"—offers electrical energy for a demand charge ranging from one dollar to 50 cents per kilowatt, and for an energy charge of from 2.2 cents to 0.5 cent per kilowatt-hour; Schedule G—"off-peak service"—for a demand charge ranging from 50 to 25 cents per kilowatt, and an energy charge from 1.25 cents to 0.47 cent per kilowatt-hour. The latter schedule, however, is applied only for large installations with not more than 100 hours of operation each week, the hours during which the energy is *not* to be used *to be designated by the Light Company*. Both schedules are based on a coal cost of $1.25 per ton.

The stipulation attached to Schedule G—that the Light Company reserves for itself the right to designate the hours during which the energy shall *not* be used—apparently makes it possible to materially reduce the charges for the energy. It is natural that

it should, because the Light Company will designate for the hours during which the energy should *not* be used, the period of its peak-load, selling the energy under Schedule G only at a time during which the capacity of the power-plant is not taxed to its maximum.

MR. J. H. VAN AERNAM:* Inasmuch as the central stations are not at present encouraging the application of industrial electric heating for a day load, because of overloaded equipment, the use of electricity in bread-baking ovens in bakeries might be worked out, which would be an off-peak load. There is little doubt that if as complete an investigation were made of this application as was done in the case of japanning ovens, electricity could compete with gas in this industry also.

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MONONGAHELA RIVER NAVIGATION

By Lieut. Col. H. W. Stickle*

1. HISTORY

The Monongahela River taps the richest bituminous coal region in the world.

The state of West Virginia alone is estimated to contain more workable coal than the whole of Great Britain. The coal mined annually, in the region immediately adjacent to the Monongahela exceeds by over 60 per cent. the annual output from the four great coal fields of South Africa in normal times.

Pittsburgh's great industrial development and her prestige as a steel center are due largely to the improvement of the Monongahela by canalization. Forty-five per cent. of the coal mined in the Monongahela valley is transported to Pittsburgh's mills by water, and these mills, now engaged principally in the manufacture of war material, depend on that 45 per cent. for their continuous operation.

The river presents the very unusual case of a large tributary with much less fall per mile than has the river into which it discharges. From Fairmont, W. Va.—where it is formed by the junction of the Tygarts Valley and West Fork—to Morgantown, W. Va., 26 miles below, the fall is 2½ feet per mile; but from Lock no. 7 at New Geneva, Pa., to Pittsburgh, a distance of 83 miles, the fall is 0.7 foot per mile, while the average fall per mile between Pittsburgh and Wheeling, 90 miles, is 11½ inches—all at low-water stages. The ordinary fluctuation of the river due to floods is 20 feet and the extreme fluctuation is 35.5 feet. The river, in the first 25 miles below Fairmont, has a number of sharp bends which, with the narrowness of the stream, place a practical limit upon the size of tows. Below Morgantown, the bed slope being small, the freshet currents are reduced in velocity and, the stream being wider, larger tows are practicable.

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Before the river was canalized navigation was limited to the portion between Brownsville, 57 miles above, and the mouth of the river at Pittsburgh. At Brownsville, connection was made with the National road leading to Cumberland and the East, During high stages boats ascended as far as Morgantown and at times even to Fairmont. Large quantities of lumber were rafted down stream.

In 1836 the Monongahela Navigation Company, under charter granted by the state of Pennsylvania, commenced work on the lower river, putting the first two locks in service in 1841, and gradually extending the system up-stream, completing Lock no. 7 in 1883. Locks 8 and 9 were built by the United States and completed in 1889, with the result that boats drawing five feet could navigate the river in low water as far up-stream as Morgantown. After much litigation extending from 1883 to 1897 the United States acquired the franchise and all the property of the Monongahela Navigation Company, and the locks were opened to free navigation on July 7, 1897. Enjoying for the first time the entire freedom from toll charges, the commerce of the river, under the influence of prosperous times which immediately followed, increased very rapidly, creating a demand for extended lockage facilities, particularly in the five lowermost locks.

2. PROJECT

The existing project provides for the improvement of the river by 15 locks and dams to afford slack-water navigation from Pittsburgh, to four miles above Fairmont, W. Va., a distance of 130 miles. Locks 1 to 7, were acquired by the United States in 1897, and Locks 8 to 15, were built by the United States prior to 1904. Increased traffic necessitated an enlargement and improvement of the locks and dams between Pittsburgh and Rices Landing, Pa. In rebuilding locks and dams from 1 to 5, the adopted standard was two parallel chambers each 56 by 360 feet, with a depth of eight feet on sills, and fixed concrete dams with movable tops; and at no. 6, a single lock of the standard dimensions with a depth of seven feet on sills. Traffic conditions have not necessitated the rebuilding of Locks 7 to 9, which have a

controlling depth of five feet on their sills. The project includes the building of Locks 10 to 15—single locks 56 by 182 feet, with a depth of seven feet on sills—and fixed concrete dams which are still adequate for the present traffic. The channel width varies from 125 feet, in pool no. 15, to 500 feet in pool no. 1.

3. OPERATION OF LOCKS

By means of flash-boards or adjustable tops on the dams, the safe navigable depth to Lock no. 7 is nine feet. Dams 1 and 5 are provided with Betwa wickets, and dams 2 and 3 with Chittenden drums. The Chittenden drums work satisfactorily when conditions are favorable.

Movable tops are not reliable and the only means of insuring a greater dependable depth would be to increase by three feet the height of the fixed dams. This would cause a loss of five days annually on account of the locks being flooded more fre-

Fig. 1. Monongahela River Lock No. 4—Down Stream.

quently and for longer periods. Lockage is now possible until the water in the upper pool is 20 feet, at which elevation boats can no longer pass under the bridges, and increasing the height of the locks would result in no benefit to navigation. The double locks practically insure uninterrupted navigation throughout the year except an average period of five days when the locks cannot be operated on account of ice and high water.

Between Pittsburgh and Rices Landing the traffic is so incessant that the ice in the pools is kept broken up to such an extent that navigation is not altogether suspended during the winter months except in exceptionally severe winters like the past one. The delay on account of the frozen river, during the past winter, was so serious that large navigation interests immediately designed various devices to combat the ice. Floods are of short duration and do not seriously interfere with navigation.

Fig. 2. Monongahela River Lock No. 4—Break in Dam Closed.

Where practicable, dams are on solid rock and, in recent construction, of concrete. In the lower reaches the dam is of rock-filled timber-crib construction, resting on the river-bed, with

upper and down stream slopes of solid timber, and with sheet-piling driven along the upper faces. The comb crests of such dams are objectionable. Ice and drift are enabled to concentrate their attacks at a single line and considerable damage has occurred requiring frequent repairs. The upper slope should be reduced to a level surface and all exposed surfaces should be of concrete, nowhere less than three feet in thickness. Scour below the cribs has occurred, endangering the dams, but failure of such dams never occurs before the partial destruction of the cribs. At Dam no. 2 is a concrete dam supported on piles with sheet-piles above and a timber crib below. A year ago this dam narrowly escaped destruction. The crib for nearly 300 feet in length was entirely destroyed, a deep scour developed and extended upward under the dam in some places to the sheet-piling above. The ability of the concrete to arch over a great distance seems to be the feature which saved the dam. The repairs to such a dam are troublesome and expensive. The new crib has a concrete top. Allegheny River Dam no. 2, which failed this month for a length of 350 feet, is of almost identical design.

In the upper dams, unless there is a rock ledge, a scour invariably develops, below, on the abutment side. Protection by riprap is necessary. Constant attention to the feature of scour, and repeated replacements of iron parts in the lock construction, constitute the chief work in the maintenance of slack water on the Monongahela.

Operating Machinery. With the exception of no. 8, all the upper locks from 7 to 15 are operated by hand. The valves and gates at Lock no. 8 are operated by power developed by 15-inch turbines—one located at each gate—which can be operated under a minimum head of three feet. The power from the turbine is transmitted through a vertical shaft, at the top of which is a worm engaged with a worm-wheel attached to a horizontal driving-shaft, on which are mounted two chain drums—one for opening and one for closing the lock-gates—and two bevel-pinions, both of which are engaged with a bevel-wheel, attached to a shaft—perpendicular to the main shaft—for raising or lowering the vertical slide-valves used for filling or emptying the lock

chamber. The valve is raised and lowered by engaging either one of the bevel-pinions on the main driving-shaft, causing the bevel-wheel and its shaft to revolve in either direction, and engaging the spur pinion on the other end of this shaft in a gear rack attached to the top of the valve-rod. The valve is balanced by counterweights.

The gates, which are of the miter type, are operated by the cross-chain method, one leaf being opened by a chain attached to the toe of the gate at the bottom and on the up-stream side, and the closing chain for the same leaf being attached in a similar position on the lower side, the chain leading to the lower drum on the opposite wall.

Compressed air is the motive power at all of the lower locks. The plant consists of a water turbine, connected by bevel-gears to an air-compressor. Air, at 100 pounds pressure, is piped from the reservoirs to all the engines operating the valves and gates, and for speed and economy of operation has given most excellent results. One stroke of the cylinder opens or closes one of the miter gates in 40 seconds, but in actual practice 60 seconds is considered advisable. The actual time consumed in closing the upper gates, emptying the chamber, and opening the lower gates or reversing the lock, is 4 minutes 13 seconds. The average time to lock a packet-boat is 10 minutes. The average time consumed by a single tow in entering and leaving the lock is 14 minutes. The average time in locking a double tow down and a double tow up is 1 hour and 30 minutes—16 lockings of 5000 tons, or 90 000 tons a day, or 32 000 000 a year, so there need be no fear for some time to come as to the lower locks being unable to accommodate the traffic.

Water-Supply. The question of water-supply is of comparatively easy solution. With the aid of flash-boards or adjustable tops it is practicable to maintain a depth of about 10 feet to near the head of the sixth pool. To maintain such a depth or to maintain even eight feet, in this stretch of the river, in case the locks were worked to their full capacity, it would be necessary to store water in the upper pools by means of flash-boards or dams of moderate height on several of the uppermost tributaries. It is

estimated that a three-foot depth, stored in the pools above Lock no. 6, will furnish sufficient water for the locks if worked to their full capacity for a period of 3½ months, and it would seldom be necessary to rely solely on this stored water.

In determining the size of the upper locks—10 to 15—the narrowness of the river for tows, the short bends, and the occasional swiftness of current, were taken into account. For a

Fig. 3. Steamer Alicia About 400 ft. Above Alicia No. 1 Coal Dock. Right Bank.

number of years coal was towed from the fourteenth pool to McKeesport—106 miles with 12 locks to pass in each direction.

The ice situation this year was about as severe in the Monongahela as ever in its history. There was a persistent gorge in pool no. 5, which could not be handled by the towboats of the river interests, and on February 11, it was decided that the District Engineer office would make an effort to dynamite the gorge at

Brownsville under the immediate supervision of Assistant Engineer Fairchild.

The lower end of the gorge began at the Bridgeport tipple ice-breakers, 0.7 mile above Lock no. 5. The ice was packed to the river-bed except in the channel where it varied from four to six feet in thickness. Farther up stream it was found to be of greater thickness and more compact until at W. Harry Brown's cofferdam, about 2000 feet above the Bridgeport tipple, it was packed about six feet below the river surface and piled to a depth of 4 to 12 feet above the river surface for a distance of 600 feet. This ridge of ice extended diagonally across the river and was wedged between the cofferdam and the mouth of a small creek on the opposite side of the river a short distance below. This ridge formed the key to the situation since it was of sufficient strength to stop the ice, piling against it from above, and by its position it protected the field of less compact ice below it. To break this key it was attacked from the down-stream side after a channel 1500 feet long had been blasted through the ice-pack below. After all shore ice possible had been loosened at the lower end of the gorge by ramming with the steamers *Swan* and *Slackwater,* blasting of the channel up-stream through the center of the ice-pack was started. Eight holes were first disposed in a semicircle of about 300 feet in diameter, the curve being convex to the direction of the current. The holes were from 12 to 18 inches in diameter, chopped through the ice or to the river-bed where necessary, with a long chisel bar. From 10 to 12 pounds of 40 per cent. dynamite per hole were tied in weighed cement sacks, lowered through the holes and suspended under the lower surface of the ice. All eight shots were exploded simultaneously. The ice was well shattered in the areas encircled by the holes and for 10 to 20 feet above each hole. After all loose ice had floated away, the *Swan* by ramming and then pulling with a snag hook, removed everything up to the solid breast. The same procedure was followed with successive series of holes until a channel 1500 feet long had been cut up-stream to the ridge forming the key. The number of holes was decreased from eight at the down-stream end of the channel to five at the up-stream end, the width correspondingly decreasing from 300 feet to 100 feet.

The width was also controlled to some extent by variation in loading of the end holes of each series. A line of five holes was then laid out diagonally, up-stream from the opened channel to the left shore opposite the mouth of the creek, and in such a position as to cut across the down-stream corner of the ice ridge. When these holes were fired, the greater part of the ice along the left bank below them was loosened and floated out and a slight movement of the ice ridge was noted. A second diagonal line of holes, paralleling the first line but about 300 feet farther up-stream, was next fired, breaking the ridge at its juncture with the left shore. The entire ridge began to move, pivoting about its right end at the W. Harry Brown cofferdam, and was quickly followed by movement of the entire gorge above that point, which at the time extended to Fredericktown, Pa. The river, at this time, was seven feet above the crest of Dam no. 5.

Moderating weather, together with softening and sinking of the ice, due to suspended silt, had started the upper end of the gorge to move and undoubtedly contributed to the success of the undertaking. In drilling blast holes it was noted that for the upper 18 inches the ice was very hard but below that point it was usually softened and honeycombed and filled with silt. After the final movement had started, the entire gorge passed a given point in three hours but it disappeared so rapidly, by shattering on the dams and sinking, that practically no ice reached Pittsburgh.

An attempt was made to use blasting-powder instead of dynamite to shatter the ice but this proved to be a failure. Five one-gallon jugs were each loaded with 10 pounds of blasting-powder and one no. 6 electric exploder. After the lead wires had been arranged in slits in the corks and sealed with soap, the jugs were suspended below the ice and the charge exploded. No appreciable shock or lifting effect was observed, probably because the action of the powder was so slow that pressure was relieved by displacement of the water.

4. ENCROACHMENTS. ACID POLLUTION

The great industrial region in and about Pittsburgh, taken as a whole, owes its supremacy to its world-renowned manufac-

tories, coal mining operations and other mineral production, which render necessary the extensive railroad and water transportation systems. The narrow bottom lands suitable for the larger manufactories and for railroad and yard transportation purposes are limited in area; so that, once established, those interests are soon hard pressed for room for expansion and every available foot is extremely valuable. To move to new locations is undesirable, costly and under existing conditions in many instances quite impracticable. Nevertheless, many manufacturing concerns have divided their plants or established new units, moving away from the original mills and factories as far as 40 or 50 miles.

When the manufacturers first occupied the river banks, they for many years engaged in filling up the lowlands to heights near to or above the flood levels. Many of the plants had been constructed too low and were, therefore, subject to the ravages of extreme freshets. In more recent years an effort has been made to build and fill above flood heights, requiring enormous quantities of mill refuse. In this the railroads have been an important factor. In order to reduce their own losses they have taken advantage of every opportunity to raise their track levels above ordinary flood danger. Thus large interests have made efforts to preserve the streams in such manner as to readily discharge the flood waters and lessen the consequent damage to properties. The value of these operations to the community as a whole cannot be overestimated.

Reverting to the river banks as nature left them, they in their neglected condition, were usually grown up with trees and bushes extending over comparatively worthless, low, bottom lands far out on the bars, rendering the rivers in their unimproved condition almost useless for navigation purposes and incapable of properly carrying the flood waters. The trees and bushes collected great drift and ice-gorges, further retarding the flow at freshet stages and causing additional bar deposits. In such rivers, filled with eddies and with landings which were poor, and expensive to maintain, water transportation was uninviting.

It is believed that the filling in of the river banks to regular lines at substantial heights has accomplished more in a general way for both the interests of navigation and for flood protection than

all other efforts, save the one fundamental improvement by locks and dams.

It has been contended that while a reasonable amount of bank filling may be an improvement both for navigation and flood purposes, the tendency of harbor line establishment at Pittsburgh has been to encroach beyond reasonable limits, reducing the streams to a width insufficient for navigation purposes. The speaker believes that this contention is not well founded. Recent calculations show that a movement of the bank line from 40 to 80 feet back from the present harbor line location in the Monongahela River at the Jones & Laughlin steel works would result in reducing the extreme flood height only 0.02 foot. There is no difficulty in choosing between (1) a wide river encouraging deposits obstructive to navigation and involving expense in their removal for the sake of a fraction of a foot in maximum flood height once in 25 or 50 years, and (2) improved navigation and channel conditions due to a more contracted river with filled banks.

"The ultimate difficulty besetting projects of river improvement is . . . caused by the fact that its beds and banks are composed of a material that is erodible."*

If bank filling to the established harbor lines is non-erodible, or if it is well protected above and below water, it should be encouraged. Encroachments beyond harbor lines, except for construction of ice-breakers, and for certain classes of terminal facilities in favorable locations, are not permitted. Laws for the protection of navigable waters are sufficient to prevent improper encroachments, or deposits which limit the navigable capacity of the streams.

The pollution of the rivers of the Pittsburgh district by the discharge into them of acid water, and the damage to boat hulls, boilers, and the ironwork of locks and dams, have been the subject of exhaustive investigation, and many recommendations have been made. The present laws for the protection of navigable waters are not generally believed to be sufficiently inclusive

*J. L. Van Ornum's "Regulation of Rivers," p. 79. 1914. McGraw-Hill Book Co., Inc., N. Y.

to justify an attempt to prevent the discharge of acid waste in liquid form or to justify prosecutions for committed acts. This situation is on the eve of remedy since the pending river and

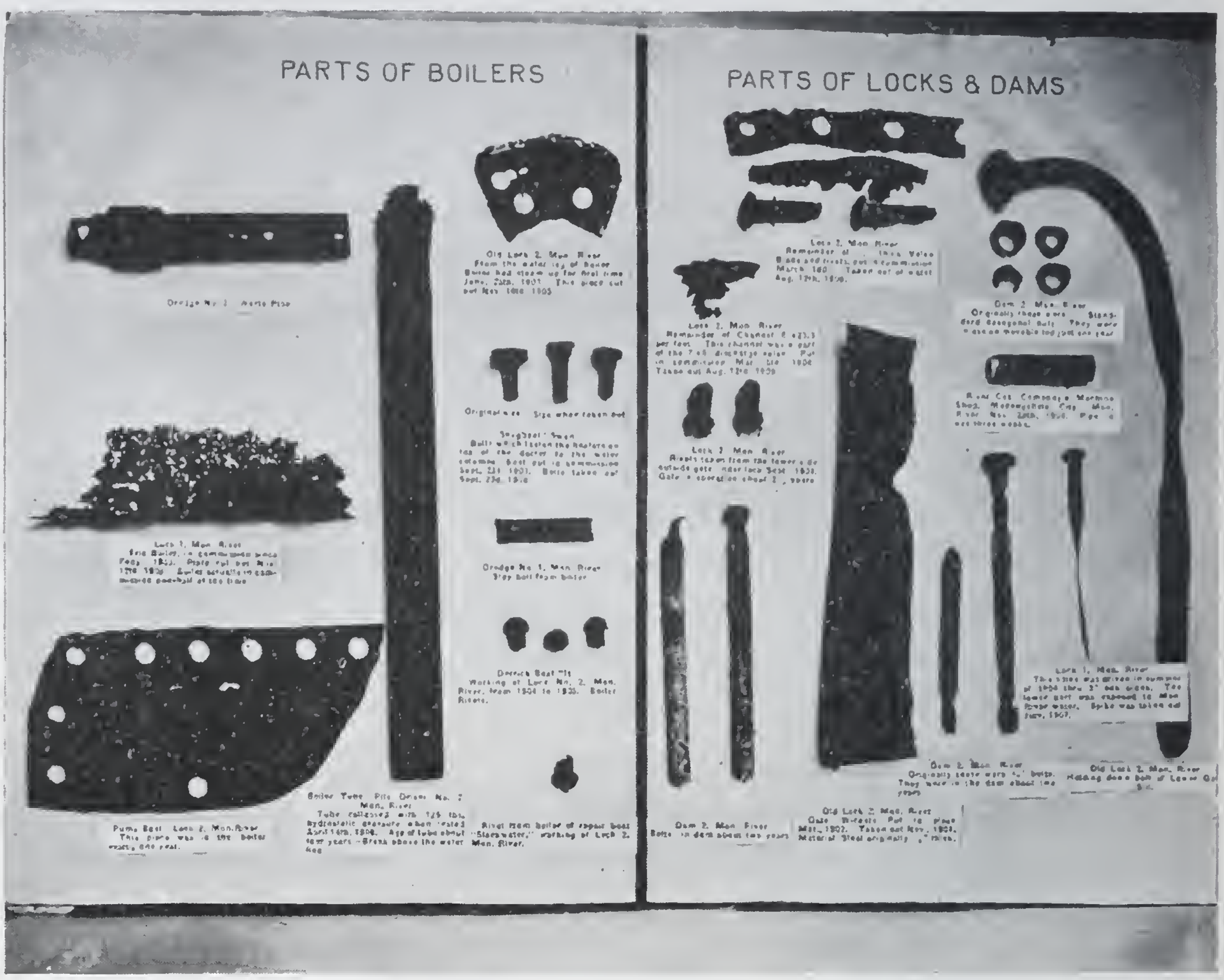


Fig. 4. Corrosion of Small Parts by Acid Water.

harbor bill contains a section reading, "That, within limits to be prescribed by the Secretary of War, it shall not be lawful to throw, discharge, or deposit, or cause, suffer, or procure to be thrown, discharged, or deposited from any source whatever any free acid or acid waste in any form, either directly or indirectly, into any navigable water of the United States or into any tributary of any navigable water," etc. The bill provides a punishment for violation and provides the method of enforcement.

In the lower 60 miles of the Monongahela River alone, hundreds of thousands of dollars are lost annually, by the Government, and the owners and operators of vessels, by the destructive action of acid water.

The Chief of Engineers states:

"The presence of acid and acid salts in the water results in deterioration to the boilers and hulls of steamboats, and damage to the submerged metal parts of the Government locks and dams.

"For use in boilers the water has to be subjected to a special treatment, the expense of which is considerable, and in spite of the treatment experience has shown that the life of the boilers is only about half what it should be. Formerly the boilers of vessels using these waters lasted, with average annual repairs, 20 years; with similar repairs they now last only 10 years. In other words, the boilers in boats employed in commerce, and those in the boats belonging to the Government and used for purposes of improvement, must be wholly renewed once in 10 years, instead of once in 20 years.

"The damage done to the Government locks and dams is extensive. The valves, gates, plates, operating chains, and all metal parts below water are corroded and eaten away by the action of the acid. Posts have been found almost eaten through after a few years' service, when in pure water they ought to be almost as good as new. While it is difficult to fix exact money value of the damage done to these works, yet, from careful estimates, excluding as far as possible all other causes, it is safe to say that the cost to the United States of deterioration due to acid in the waters is not less than $25,000 a year on the Monongahela, and not less than $32,000 a year on the Ohio."

The proposed legislation will provide a means to stop this constant damage, and should be welcomed by all industrial and navigation interests along the river.

5. AMOUNT AND CHARACTER OF COMMERCE

The Pittsburgh harbor commerce exceeds that of any other inland river port of the United States. For 1917, the Pittsburgh harbor tonnage was about 14 000 000 tons. At no time in the past has the traffic by water in the vicinity of Pittsburgh approached its present volume; neither has the prospect at any time for extension of water-borne commerce been so promising.

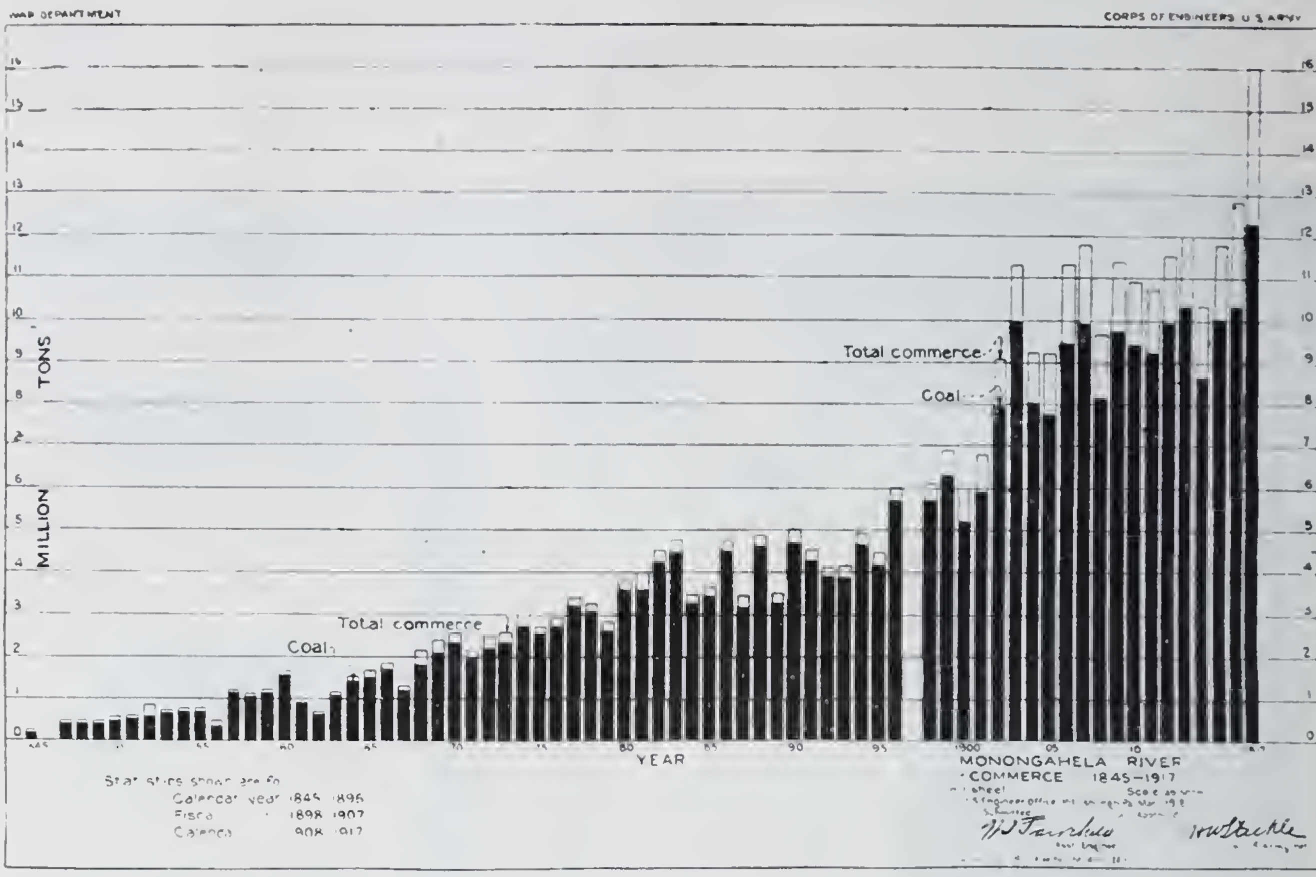


Fig. 5. Growth in Commerce on Monongahela River, 1845-1917.

The chart (Fig. 5) shows the growth in commerce on the Monongahela River from 1845 to 1917. This commerce gradually increased, with annual fluctuations, from 250 000 tons in 1845 to 6 900 000 tons in 1900. Then it began to increase more rapidly reaching approximately 12 700 000 tons in 1916, practically doubling in 15 years. The tonnage for 1917 is 16 000 000. In spite of the great decrease in coal shipments from the Pittsburgh district to ports along the Ohio and Mississippi Rivers, and the increased shipments by rail to meet the demands of the eastern markets, the consumption of coal in the Pittsburgh district has been so great as to show an increase in tonnage on the Monongahela River. This remarkable increase in tonnage was made notwithstanding the fact that the railroads parallel the river on both sides. Since the canalization of the river the low land has been gradually filled with waste material from the furnaces and is now used for building sites for the extension of plants or for new works. New manufacturing towns have been built along its banks, and concerns of material reputation, such as the Carnegie Steel Company, Jones & Laughlin Steel Company, American

Steel & Wire Company and others producing immense quantities of raw and semi-finished materials—as pig-iron, billets, steel rails, spikes, pipe, bolts, rods, plate-glass, heavy machinery, armor plate, wire, brick, tile, chain, tools, sulphuric acid, boiler tubes, etc.—are preparing to ship by river.

To give an idea of the prospective commerce it may be stated that the by-product plant of the Carnegie Steel Company at Clairton, when completed, will require over 8 000 000 tons of fuel a year, and this coal will be moved from mines in the various pools by the Company's own boats which are now under construction. In addition to the shipment of coal by river, to the plant at Clairton, approximately a million tons a year of beehive coke will be handled by the Carnegie Steel Company's boats from its coke plant in the sixth pool to Monongahela valley plants and blast-furnace plants on the Ohio River. The Company also contemplates the handling of raw material and semi-finished materials such as pig-iron, etc.—amounting to 3 000 000 tons a year—between the various plants.

The towing of coal constitutes by far the greater part of the business of the river. The dimensions of the barges now being built by the principal firms are governed by the dimensions of the standard lock chamber—56 by 360 feet. These barges are 175 feet long and 26 feet wide and may be loaded to a draft of eight feet. Four of these barges, or three and a towboat, fill a chamber. The bulk of the coal is mined in the fourth, fifth and sixth pools and is brought down from the mines in single tows of 2450 tons and double tows of 5350 tons.

Formerly, at double locks, only one chamber was ordinarily in use, the other being held in reserve against accident and emergency. The tows were then made up so that each lockful had its own towboat. At present both chambers are used and each tow is made up of two lockfuls, which are sent through simultaneously if both chambers happen to be available; otherwise, in succession. One of the largest operating companies states that by double locking it saves 33 per cent. of the former cost of transportation.

The best information available is to the effect that the bulk of the coal tonnage—which has now reached over 12 000 000

tons—is moved at rates varying from one-eighth to one-third of the corresponding rail rates.

Assured of practically uninterrupted navigation,. owners located steel-mills and various other industries along the river, primarily to take advantage of cheaper coal and ultimately to be in a position to ship their products by water. Some of these interests acquired coal lands, built towboats and steel barges, and installed improved unloading devices at their plants. This enabled them to materially reduce their fuel costs and insured them against a coal shortage. Certain companies have, in addition, acquired large tracts of coal lands, contracted for a large number of steel barges and towboats, and undertaken the construction of by-product plants. From Pittsburgh to Charleroi, Pa., the banks of the river are now lined with steel-mills, and other manufacturing establishments.

The rich coal fields tributary to the sixth pool are now being developed. Mines have been opened; tipples have been built; and one of the largest steel concerns in the country, not slow to recognize not only the economy but the dependability in transporting coal and coke by river, has purchased immense tracts of coal land and will ship over 3 500 000 tons a year from its mines in the sixth pool. This company, heretofore, depended almost entirely on coal shipped by rail. Three concerns now control deposits of a billion tons or more in the sixth pool, and will ship by river. On account of these developments a second standard lock should be built, not because the existing lock is of insufficient capacity to do the business, but for the reason that the disablement of the lock might result in the stoppage of works giving employment to thousands of men. It is practically impossible, in this district, owing to lack of room and handling facilities, to store fuel in large quantities.

It is quite evident, if any extensive coal business is to be developed above pool 6, that Locks 7, 8 and 9 should be rebuilt, following the general design and dimensions of the lower locks. These locks are single and are only 160 by 50 feet, with a depth of five feet on the sills. With flash-boards, the depth could be increased to eight feet, but at best they could be made available only for specially constructed barges, 160 feet long and 24 feet

wide. With seven-foot draft, two of these barges would have a combined capacity of about 1350 tons, and if a steamer had four such barges in tow the total would be 2700 tons, requiring three lockages. Compare this with a standard lock, quite capable of passing this tonnage through in one lockage. In computing the cost of transportation on rivers, the time spent at locks is a factor of considerable importance.

Packets. For a long time the packets did a good business. Before the railroad lines were completed to Morgantown six large packets operated between there and Pittsburgh, and after the completion of the locks between Morgantown and Fairmont there was considerable business on the upper river. On account of mismanagement, railroad competition and other causes, the packet business gradually declined and finally the *Valley Gem* while in the possession of the United States Marshal made a run on the bank and was frozen out of the game by the ice-gorge at Morgantown, W. Va., on February 9, 1918.

6. TERMINALS

The advantage of lower freight rates by water, as compared with railways, will not alone satisfy the demands of modern business. At several landings the packet companies had wharf boats but it was a frequent occurrence to unload valuable freight on the bank, where it was liable to be stolen, or destroyed by high water. Care of valuable goods is an important factor in the cost of transportation and if the packet trade is to be profitably revived it must be operated under modern business methods. River terminals for freight must be provided at prominent landings and these must be in all respects equal to the railway terminals—weather-proof and accessible at all times to wagons and motor trucks. With this done along the river the cost of actual movement between given points, if cheaper by river than by rail, will be sure to attract the attention of the business man, and a fair share of general business will seek the river.

For a very extensive business at river terminals, especially where freight must be transferred from boats to railroads for

points in the interior, proper track arrangement must be provided. The railroads reach all parts of the interior and it is a matter of great concern to the general shipper that equitable and legal arrangements be made for interchange of traffic between the river and the railways.

The present method of appropriation for river and harbor improvement is a development from what could, not inappropriately, be designated a "pork barrel" method, when congressmen exchanged courtesies in voting money for various districts. At present, however, no project will be considered for appropriation unless it has the recommendation of the Board of Engineers for Rivers and Harbors, so, no matter what other criticism may properly fall upon appropriations for river improvement, the cry of "pork" is unjustified. For the past few years great emphasis has been placed upon the question of terminal facilities. In the pending river and harbor bill appears the following:

> "The Secretary of War is hereby requested to investigate and submit to Congress on or before the first Monday in December, nineteen hundred and eighteen, a report showing (a) the status of water terminals at cities and towns along the Ohio River between Pittsburgh and Cairo, inclusive, and whether owned by municipalities or some other public agency, and whether the same are satisfactory as to location, construction, and equipment; (b) the names of cities and towns where an interchange of traffic exists between the water transportation lines and the railroads; (c) a list of the water transportation lines existing and proposed on the Ohio River with a description of the number and type of boats in operation and under construction or to be constructed and as to whether the same are appropriate and suitable for the traffic; (d) the names of cities and towns where no adequate public terminals exist, together with a statement of any prospective plans for water terminals and the status of same: (e) any recommendation for the development of transportation on such river."

This is probably a step in the right direction, but in my humble opinion they have the cart before the horse as far as the Ohio River is concerned. The Ohio can never amount to much until it becomes a dependable river, not only for 50 or 100 miles but for the entire distance from Pittsburgh to its junction with the Mississippi. Municipalities and private individuals will construct terminals only when they can definitely figure a reasonable return on the investment. They are going to have no

sentimental hysteria about the sacred importance of terminals on an unused river. Some day there will be needed a capacious and modern terminal development to transfer freight from river to rail for the East, and vice versa. This is essential for a reasonably full fruition of the Ohio improvement. Who supposes the railroads will experience a change of heart, and dig into their pockets to pay for such a terminal? What boat line would consider such an investment in view of the lack of law to keep the railroads in line? What private individual can see anything in this for him? Such a terminal development, coupled with proper railroad laws, is river improvement just as much as dam construction, and dredging; yet the United States has not adopted any policy for terminal construction, and no such policy is in view. The trouble with the Ohio River is that the improvement is uncompleted and will remain uncompleted for many years, under the present system of comparatively small annual appropriations.

The present Ohio River project was initiated by the act of March 3, 1879, which made appropriation for the Davis Island Dam. That was 39 years ago and, due solely to lack of appropriation, the project is yet far from being completed. Imagine any corporation adopting a plan of improvement for any purpose and dragging the allotments out in such a fashion!

The *Pittsburgh Dispatch* of February 4, 1918, contained an editorial, entitled "The Ohio Appropriation", which well expresses the situation as to river terminals; having under consideration the alleged suspension of action by the Rivers and Harbors Committee while awaiting information as to what the people were willing to do toward providing terminals. This editorial reads in part:

"Yet, while it is unreasonable to blame the river towns for failing to provide terminals for a traffic possible only after the Government has completed the improvement, the committee is right in warning valley communities that the Government should not be asked to put millions into rivers that will not be properly used. A definite program should be undertaken to assure that when the Ohio is canalized the river communities will be in shape to utilize it immediately, not by the primitive wharfboat and landing methods, but by the most modern means, capable of loading and unloading bulk rather than packages, and providing prompt transfer where necessary to or from the rails. Unfortunately, many

places have permitted their wharfage to be alienated, but most towns have enough left, or that may be recovered, for the construction of modern terminals. If the committee's action stimulates public interest in planning for such facilities, to be ready as soon as the river is completed, it will be a good thing."

7. RAILROAD COMPETITION

The commerce on the Monongahela has increased by bounds, in spite of the fact that its course is paralleled by railroads on either side. It is estimated that there is, on the average, a saving of 35 cents on a ton of coal as compared with transportation by rail. This would mean a saving to somebody of $4 200 000 in 1917, which would more than replace every lock and dam on the Monongahela every three years, after deducting operating and maintenance charges. The value of this improvement is therefore unchallenged. This is a special condition and most river improvements in this country could not stand a close mathematical test so well.

Rail rates are evidently so made that traffic which might want to use waterways, finds the rail route cheaper in the long run. Rates to and along competitive waterways are kept low, and railroads are enabled to keep most of the traffic, and make it up by higher rates to interior points. Exact comparative rates bearing on this matter are to be found in an address by Senator Ransdell in the United States Senate on September 29, 1917.* Rates are invariably much higher to inland points—in some instances more than three times as much. This situation is the prime reason why commerce on the rivers cannot hold its own. Eliminate this evil—which is manifestly unfair to the interior points but which constitutes for communities on navigable waterways an arbitrary rail rebate—and the strategic advantage which waterway communities have by virtue of their location would naturally manifest itself. If rates per mile to interior points were to be made the same as for equal distances between river points, the railroad experts would certainly scream; but why should not railroad rates be based upon mileage, class, carload lots or less.

*Congressional Record, Sept. 29, 1917, v. 55, pp. 8236-8248.

and terminal charges, and be kept independent of extraneous matters such as competition with rates by water? The improvement of rivers in this country has been justified by the reduction of freight rates on railroads, and every river lock is a monument to this fact: but why should rivers be improved for any such purpose? The same effect would be produced by a legislative enactment of great simplicity.

Hon. James A. Frear submitted a minority report on the pending river and harbor appropriation bill, which is published in House Report 350, pt. 2, Sixty-fifth Congress, second session. This report attacks the bill generally, but contains some concise statements of interest on this subject. He says (page 14):

> "Constructive efforts to promote commerce on our inland waterways involves fixing railway rates to prevent roads from underbidding water transportation companies on heavy bulk freight.
>
> "Improved terminals, modern devices for loading and unloading, coordination of railway and waterway traffic, both physically and through joint rates, are all desirable, but are more ornamental than useful under existing conditions . . .
>
> "A first requirement is to control railroad rates so waterway traffic can live. Without that change in transportation conditions, failure is inevitable."

The speaker is in full agreement with the quoted statements.

8. THE MONONGAHELA AS A WAR FACTOR

Last summer the repairs and maintenance of the lower six locks and dams on the Monongahela were declared to be important for purposes of national defense, thus placing this work in the same class as shipbuilding and cantonment construction.

Germany long since knew that improved waterways were vital for purposes of war. Thus we see canal construction and river and harbor improvement going on there, at this time, under substantial appropriations. Both railroad and water rates and the commodities each system must carry are there determined after the closest study for the benefit of the country as a whole, and not with the prime object of assuring substantial dividends for watered railroad stock.

The Monongahela is unique in its present usefulness. I know of no other interior stream in this country which is giving the same direct defense and service in this war. With its peculiar situation the Monongahela is doing its part, but throughout the United States—due to uncompleted projects on account of lack of funds; due to lack of terminals; and due to unfair railroad rates—the waterways have not given their fullest and much needed contribution to the transportation requirements of the war.

This is a war fundamentally of transportation, and improved and fully used waterways would have been worth billions to us had they been available. But they were not ready and are not in use. Consequently, to save money for the immediate purposes of the war, the river and harbor bill is pruned to the quick. If the war were sure to be terminated within the next year, this would probably be good policy, but, if a man had neglected to roof his house until the rains set in and he had no reason to doubt but that the rain might continue for 5, 10, or 15 years; if he were a man of intelligence he would proceed to get some kind of a roof on his house regardless of the difficulty and expense connected with the undertaking, and would still be happy if the sun should happen to shine the next day after completion.

The Pittsburgh district is doing its full share in the conduct of the war. How much its industrial activities—gigantic as they have been the past year—would have been accelerated had a usable canal been in operation for the transportation of ore, coal, and other materials between Pittsburgh and Lake Erie!

All the difficulties in the way of water transportation will eventually be removed. I am confident of that. We will come to realize that movement of heavy commodities by water is an economic necessity and that water transportation routes are a vital necessity in time of war, and indeed a potent insurance against war. Our millions will be devoted to preparedness against war and our billions, and a million lives, will be saved.

DISCUSSION

Col. T. P. Roberts:* Colonel Stickle's address covers the subject of the Monongahela River very completely, and the occasion of its reading is quite appropriate for discussing the merits of water transportation in general—for there is nothing to prevent the magnificent results obtained on that river being duplicated wherever locks of adequate size can be supplied with the requisite volume of water. Of all the problems which now call for national attention, none is more important than relieving the railways of low-grade freight—such as coal, iron ores, building materials, etc.—and handing over such commodities to boats. This is done at Newport News, for instance—for it would be absurd to think of hauling by rail, via New York City, West Virginia coal which was intended for Boston. In Pittsburgh, however, our people patiently submit to the passage of trains a half mile long through the city—to the extent of many millions of tons annually of coal and coke, intended for use in Cleveland, Toledo, Chicago, Milwaukee, Detroit and Duluth. This incessant railroad traffic is, in many ways, a positive detriment to business interests here, and it is this business, and none other, that has led finally to freight congestions—worse here than elsewhere in the country. This local trouble should not be charged to the railroads, but it surely never would have arisen if the projected 2000-ton boat canal had been constructed, connecting the Ohio River with Lake Erie.

Affairs with the Ohio River are now in the transition stage and the necessity of a better understanding of the needs of our future commercial development, both by rail and by river, is attracting the attention of our national statesmen. Two hundred miles of the river are improved, and in two more years we will have a reliable, almost continuous waterway connection with Cincinnati, 468 miles down the river. At this very time the necessity of waterway terminals, or warehouses, at the 40 prominent cities and towns along the river, from which river freight can be distributed by rail to interior points, is engaging the attention of some of the civic authorities.

*Engineer, U. S. Engineer Office, Pittsburgh.

It is recognized by the friends of the river that no traffic commensurate with the cost of the river improvement, is possible until adequate terminals are provided.

Mr. W. E. Snyder:* Colonel Stickle has just mentioned the use of canals and improved waterways in Germany. I have spoken before, in this Society, regarding some of the features of German conditions which I have had the privilege of seeing in considerable detail, but I will repeat part of that discussion here because of its importance, even though some of you may have heard it before. I believe we are sufficiently broad-gaged to recognize merit where merit exists, regardless of present war conditions.

In 1913 I had the opportunity to go in a small gasoline launch with an experienced German shipping man, entirely around the harbor at Hamburg. I later had the opportunity of going over the slips provided in the towns of Duisburg and Ruhrort and at the works at Rhinehausen on the Rhine, and it is astonishing to any one familiar only with conditions as they are in this country, to see what wonderful provision has been made in the way of terminal and dock facilities to utilize the benefits of water transportation so far as possible—this in the comparatively small manufacturing towns, as well as in the large cities. While I have not had any special experience in such engineering work as is discussed in the paper this evening, I do know good work when I see it, and recognize the importance of those large and very complete developments, which I have just mentioned having seen in these German cities, as being of great advantage in the transportation of all commodities.

Similar conditions were apparent in every part of Germany. Improved barge canals interlace rivers, and the barges used for transportation were well adapted to both canals and rivers for very long distances.

In the eastern part of German Silesia, not far from the town of Kattowitz, I was told that iron and steel manufacturing plants there had furnished a great many installations of large hydraulic and water-supply pipe systems for the western coasts of North

*Mechanical Engineer, American Steel & Wire Co., Pittsburgh.

and South America, from Alaska south to Chile; notably a great deal of the water-supply pipe for Los Angeles, Cal. All of this—notwithstanding the seeming isolation of this manufacturing district in German Silesia—because of transportation by canal and river, sea and ocean, from the point of manufacture to the point of consumption.

The great difficulty in this country is to have the legislative bodies which control the distribution of funds applicable to river and canal improvement, recognize such matters without having actually been in Germany to observe what has been done, and its result. It is because of the ignorance and indifference of the general public regarding such matters that no systematic program of river improvement and canal construction was ever adopted, and this ignorance and indifference of the public is always desired by professional politicians, as it enables them to make each case of necessary waterway improvement a law unto itself, with some surrounding circumstances which can be twisted to benefit their own personal plans or fortunes.

Papers and discussions of the kind we have had this evening are what is required to enlighten people as to the importance of the whole question of improved waterway transportation; and the pity of it is that such discussions must be confined to a comparatively small body of men and never reach the thinking public, many of whom are indifferent only because they have never had the opportunity to know the facts.

Such a complete breakdown of railroad transportation as occurred during the past winter emphasizes the importance of making more use of the waterways, both natural and artificial; and with the lessons of the past winter in mind, it may not be too much to hope that this whole subject of waterway improvement will be treated by Congress according to the best engineering requirements, and in a manner commensurate with its importance.

Mr. F. L. Egan:* I feel that we owe Colonel Stickle a debt of gratitude for this paper, and it comes at an opportune time, as due to war conditions we are all interested in transportation.

*Draftsman, Carnegie Steel Co., Pittsburgh.

Would Colonel Stickle give us some information, as to why the stern-wheel towboat is in almost universal use on our American rivers, while the side-wheel paddle; single, double, and triple screws; chains; and in fact all known methods of propulsion, except the water jet and the stern paddle wheel, are used on European rivers?

The difference in the method of towing, can hardly explain it because, of late years, practice on the Nile has been to tow at the head according to our American practice. In fact this method of towing was, I understand, introduced on that river by an American builder of stern-wheel boats and they continue to use it, but with twin- or triple-screw boats.

For instance the steamer *Wai,* built by Dunsmuir & Jackson, of Glasgow, Scotland, for the British Government, and for river service on the Bombay coast of India, replaced a stern-wheel boat. The *Wai* was 90 feet long, 20 feet in beam, 3.25 feet full-load draft, with triple-expansion, three-cylinder engine; $9 \times 14\frac{1}{2} \times 25$ inches, with 10-inch stroke; 300 r.p.m.; Joy valve-gear, and three-bladed screws 2.5 feet in diameter. The single engine stands across the hull, and each cylinder drives its own shaft and screw, although the three shafts are coupled with double parallel rods similarly to a locomotive. This Scotch company claims to have similar boats running successfully on as low as 16 inches draft.

One of our local companies, building river boats, has a number of stern-wheel boats in service on a South American river, and on the same river, using the same method of towing, the British Government is using twin and triple screws, and has done so very successfully for several years.

Col. H. W. Stickle: It is very largely due to the way you start. The stern-wheel type was developed from a boat which first navigated over a heavy dew, as the old expression had it, and our waterways have been kept in that condition very largely. It makes a good working boat but you can not put a tow-line on the stern of a stern-wheel boat and operate it. On the Nile there are a great many of these twin-screw boats and I think you will find in the near future a number of such boats on these rivers.

As we secure additional depth I think you will find more of those boats coming in. I myself have been used to a different kind of towing from what I have seen here. I had always seen boats towed with a line. That seems to be ordinarily the most satisfactory method. Of course, in shallow streams where they have to turn quickly, I suppose the stern-wheel boat serves the purpose just as well, or better.

Col. T. P. Roberts: Regarding stern-wheel boats, I would like to add several points which appear to be misunderstood by eastern and lake boat builders, for they ridicule Pittsburgh boat builders most unmercifully. Our boats are equipped with several balanced rudders, and coupled together to act as a unit so that when the steamer backs its wheel it throws against the rudders nearly the whole weight of water—moving 10 to 12 miles per hour, if need be—which has been put in motion by the wheel. The force possible of development in this manner is equivalent to from one hundred to more than five hundred horsepower, dependent upon the length and depth of the wheel blades. Practically all steering of stern-wheel boats is done by backing the wheel, ordinarily for but very brief periods, and with, apparently, but little loss of motion of the fleets.

A tunnel boat, with rudders forward of the propellers, has this power only to a limited extent, because there is not room enough available in the tunnels to present the really enormous area of rudder surface found on stern-wheel boats. Balanced rudders, 22 to 35 feet long, extend forward of the rudder-posts, and back to and beneath the stern wheel to nearly its half diameter —the wheel blades almost touching them. The forward sections of the rudders are shaped to clear the shear of the steamer's hull, so as to reduce water "slip" to a minimum. When balanced rudders were introduced (about 1870, no patent having been taken out), it was soon discovered that the efficiency of towboats on the Ohio was at once more than doubled—and this, too, without requiring any change in the engine power. It was something really wonderful, and of vast importance to Pittsburgh, for it made possible transportation of bulky commodities like coal at

the cheapest rate known to the world. It is quite possible—and it has been done hundreds of times with second- or third-rate steamers of, say 700 to 900 horsepower—to move from 15 000 to 20 000 tons of coal to Cincinnati (468 miles), and to return with empty or partially loaded boats, in seven days; provided, of course, that there should be several feet spare depth in the channel. There is little difference between the time of moving up and the time of moving down, at such periods. From Louisville or Cairo to New Orleans, on medium high stages, large towboats have frequently moved 45 000 tons and, in the case of Pittsburgh's largest towboat, 60 000 tons of coal, bringing back empties or sometimes partially loaded barges. The largest tow included fifty-six 1000-ton boats, besides several fuel barges. The length of tow was 1130 feet, the width 312 feet and the area of the fleet about eight acres. With good river conditions the round trip—about 2800 miles—from Louisville to New Orleans and return, was sometimes made in less than 23 days. Imagine a propeller handling such fleets; or imagine it pulling them with a cable.

It cannot be denied that for much of the time it is practicable, on the Mississippi between St. Louis and New Orleans, to do as is done on the Danube and the Rhine—viz. to pull boats with cables—but there are times on the Mississippi when it is necessary to stop the fleets and tie up to trees, and it is in such emergencies that the stern-wheel boat displays itself to best advantage. When moving down stream with a current of five miles an hour Pittsburgh towboats can check their fleets and flank them in to the shore with perfect safety. A steamer not able to do this, at any time the necessity arises, is not safe for heavy fleets on the Mississippi.

It is safe to say that the volume of traffic, as now conducted through the locks on the Monongahela River, would be impossible with either a cable-towing system or with pushing steamers not provided with balanced rudders. Steamers so provided frequently enter a lock chamber with 3000 tons in tow and pass out without resorting to the use of the power capstans on the lock walls. An entire year elapsed at one lock in frequent day and night use

during which period the lock capstans were never used to assist the passage of vessels through the chambers.

Capt. John L. Kerr:* After a number of years' experience as a master and pilot on the Ohio and Monongahela Rivers and on boats of every type and size which have been used and are at present in operation on these rivers, my opinion is that it is not practical to tow behind a steamer on a tow-line on account of the current, the narrowness and crookedness of the river and the channel, and the necessity of stopping frequently on account of bad weather, smoke, etc. I do not believe screw propellers are practical in these waters except by towing ahead at a dead slack-water stage, and then only on a straight run; since in working around the locks—a matter which occurrs every two or three hours—they could not be handled, with their tow, nearly as rapidly nor as accurately as a stern-wheel steamboat. A stern-wheel steamboat will back around or twist, either loose or with a tow, very much more quickly than any screw boat that has ever been constructed, and I can not but conclude that a screw propeller with tow would always be in the way of the more rapidly handled stern-wheel boat; and when working about the locks at the time of rises in the river, I do not believe that the screw-propelled boat with tow would be altogether safe; as there is always a certain danger of going around guide walls and over dams unless the steamers are of such a type as can be handled quickly and properly. I do not think the screw propeller will meet the requirements, and I wish to heartily endorse what Colonel Roberts has said on this subject.

Mr. R. A. Cummings:† I wish to endorse the statements of the two previous speakers, relating to the facility with which the large stern-wheel, balanced-rudder steamboats handle tows of large coal boats.

This type of boat has survived practical and technical criticism, for large tows on shallow rivers. The principal function

*Captain and River Pilot, Monongahela and Ohio Rivers, between Pittsburgh and Louisville.

†Civil Engineer, Pittsburgh.

of a towboat in down-stream traffic is to control the movement of the tow, in passing bends, by flanking. By reason of the large balanced rudder used on the stern-wheel boats, these can develop much more flanking power when backing than it is possible to obtain otherwise. Adoption of the side-wheel boat used on European rivers is doubtless due to the method generally adopted there of towing the fleet astern of the towboat, which is not applicable to our western waterways, largely on account of their physical condition and the fact that a crew would be necessary for each barge.

The use of the tunnel boat has been limited. Tests were conducted on the Kanawha River seven years ago, with very encouraging results, and doubtless this type of boat will come more and more into use as rivers become canalized.

I was very much impressed by Colonel Stickle's address, and his reference to the fact that the concrete work in the locks has not deteriorated by reason of the acid condition of the river, but that the steel in the lock-gates, valves, etc., has been destroyed in a very short time. I know that the steel hulls of certain boats operating on the river have been deeply pitted. This matter, as a feature of the concrete boat, should command sympathetic consideration.

Another matter with which I had to deal to-day related to the silting of the rivers. A floating dry-dock project was under consideration, and it was stated that, in addition to the limiting natural depth of water, trouble might be anticipated from silting under the dock, thus interfering with its operation except at considerable expense by dredging. Personally, I am opposed to this view, for the reason that trouble from silting has rarely been experienced on the Monongahela River. Although this stream is canalized for 120 miles above Pittsburgh, experience has not indicated trouble from silting.

For this reason it is my opinion that the argument against the use of the floating dry-dock is negligible. It is my impression that the silting which occurs on the rivers in this vicinity is largely confined to the Allegheny and Ohio, and I should appreciate it if Colonel Stickle will tell us whether or not this is true.

COL. H. W. STICKLE: It is undoubtedly true that when the ice went out in the Allegheny River more silt was found than ever before. Such silting is very small in quantity and very easy to control. As a general proposition the pools will not silt up to any noticeable extent. A very small amount of dredging in the pools keeps that all right. Of course, with exceptional floods, and the ice going down carrying silt as was the case this year, we get some silt. It is largely in the Allegheny and the Ohio; none that I know of in the Monongahela this year.

MR. F. L. EGAN: In regard to the maneuvering ability of the two types I would refer you to pages 4-5 of "Experimental Towboats", 1914. House Document no. 857, v. 27, Sixty-third Congress, second session. Captain Martin has had many years of experience, and his statement regarding the *Scott* agrees with results obtained with screw boats on the Nile and on South American rivers. A parallel case is found in the ferry-boats in use around New York City. Until the year 1889 the universal type of ferry-boat was a double-ended hull, with side wheels of large diameter placed at or very near the middle of the hull and driven by the American type of beam-engine. These boats are required to make a speed of about 12 miles an hour in service. The first screw boat placed in this service was the *Bergen,* described in the *Transactions* of the American Society of Mechanical Engineers, v. 11, p. 372. The novel feature of the *Bergen* is the use of two screws—each eight feet in diameter and 8.9 pitch—one at the bow and one at the stern, on a single shaft running the entire length of the boat, driven by a single-expansion engine $18\frac{1}{2}\times27\times42$ inches, with 24-inch stroke; running 142 to 162 r.p.m., with 150 pounds of steam and five-eighths cut-off in each cylinder. A series of very complete tests of the *Bergen* in comparison with two side-wheel boats of equal tonnage demonstrated the superior operating and maneuvering ability of the screw boat and marked the beginning of the end of paddle boats in this service.

The double-ended, screw-type ferry-boat has developed until engineers of one of our great railway systems, which operates

a number of these ferry-boats, demonstrated that one of the later type boats can be turned through a complete revolution without making way in any direction; that is, it can be pivoted about a central point, and this expediently. This is certainly a maneuvering ability of close to one hundred per cent. and we should be able to increase the flanking and turning ability of our river towboats by taking advantage of the greater thrust of properly designed screws, and the greater rudder area possible in such a design. The balanced rudder is just as easily applied to the screw boat as to the stern wheel and with tunnel screws it is easy to add the so-called monkey rudders, and secure even greater rudder area than is practicable on a stern-wheel boat of the same dimensions.

A comparison of the two methods of propulsion, applied to a 34-foot beam towboat, according to data obtained in one of the United States Government testing tanks, gives the results indicated in Fig. 6-7.

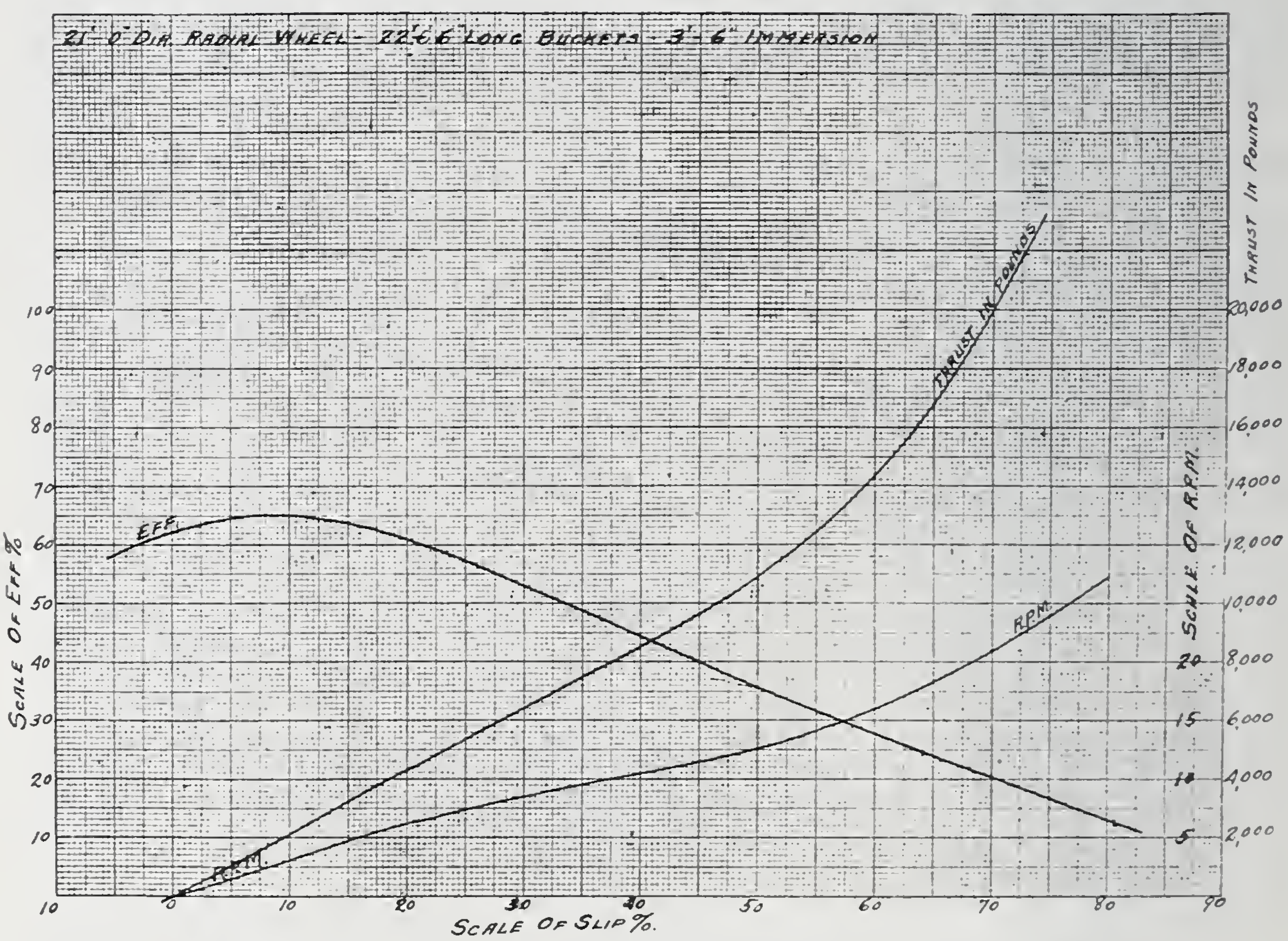


Fig. 6. Efficiency Curve of Screw Propeller.

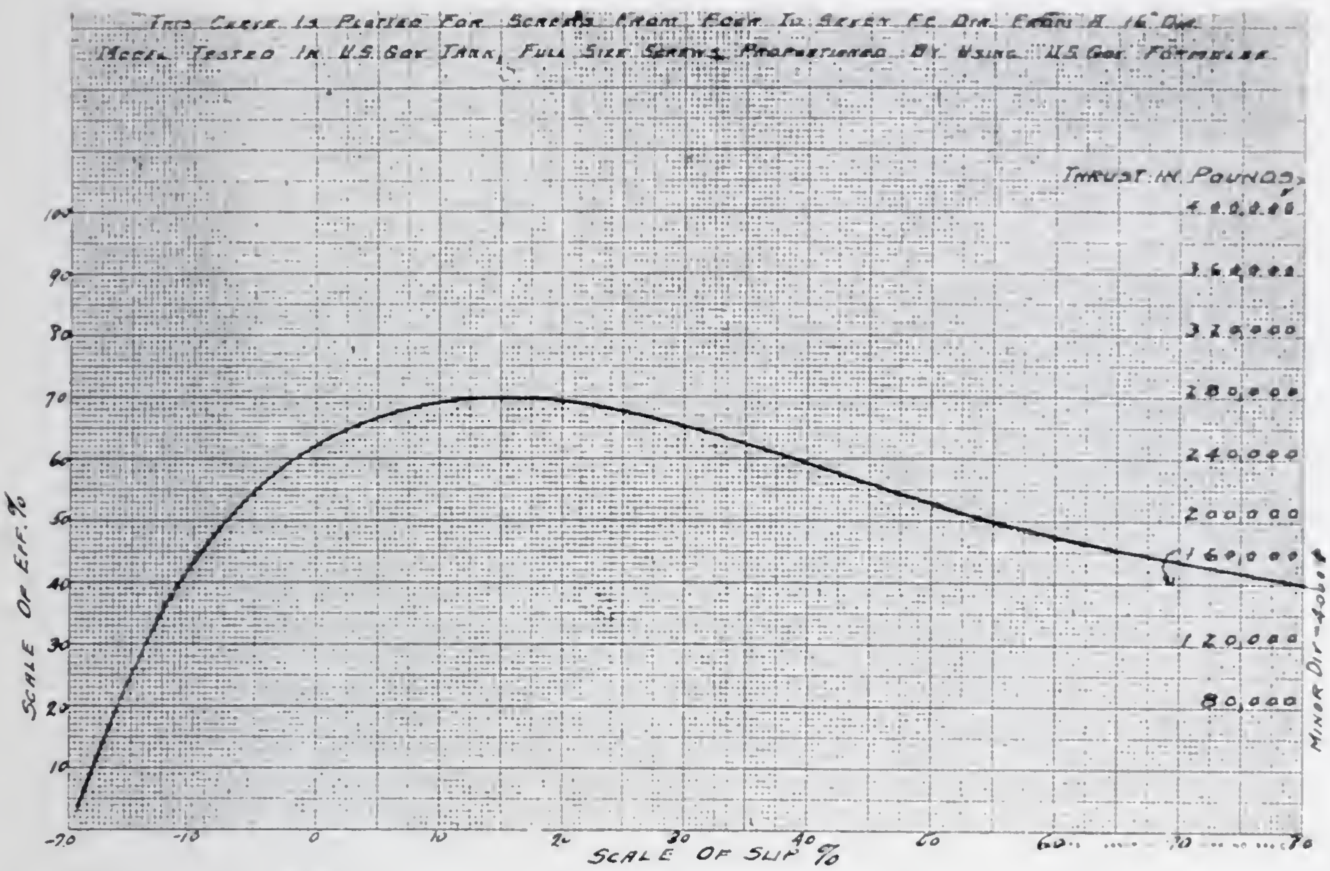


Fig. 7. Efficiency Curve of 21-Foot Diameter Radial Wheel.

It is interesting to note that the United States Engineers in their report—pages 2-3 of "Experimental Towboats" mentioned above—recommend the building of two boats, one twin-screw tunnel type, with water-tube boilers or the combination type; and one stern-wheel boat, feathering wheel type, with internally fired boilers of the return-tube type, or the combination fire- and water-tube type.

In such ice conditions as we had in the Pittsburgh and upper pools the past winter, a feathering stern wheel would be a very expensive thing, if not a complete failure, while twin-screws, properly designed, and installed in protected tunnels, would give higher propelling and maneuvering efficiency, and would probably stand the ice as well as any mechanical device that is applicable.

SIMPLIFICATION OF RIVETED JOINT DESIGN

By H. A. S. Howarth.*

This analysis of riveted joints is a study of the method of calculating joint efficiencies published in the Report of the Boiler Code Committee of the American Society of Mechanical Engineers, and elsewhere. It has resulted in the simplification and shortening of the calculations necessary to determine the properties of the joint. Its results are tabulated for the joints listed below. The analysis is given in detail in the appendix.

1. Lap-joints. Pitches the same in all rows.
2. Butt-joints, with two straps of equal width. Pitches the same in all rows.
3. Double-riveted butt-joints, with two straps of unequal width.
4. Triple-riveted butt-joints, with two straps of unequal width.
5. Quadruple-riveted butt-joints, with two straps of unequal width.

The main idea involved is this: By comparing the ratio of rivet diameter to plate thickness with certain predetermined constants, we can eliminate from consideration all but two of the possible ways in which a given joint may fail. For the simpler joints but little time is saved, but for complex ones we may save a large percentage of the work and eliminate many chances of error.

A rivet, whose diameter is d and whose sectional area is $a = \frac{\pi d^2}{4}$, may fail by shearing or by compression. The area it opposes to shearing is either a or 2a, depending on whether the action of the plates is such as to cause it to shear at one or at two places. The area opposed to compression or crushing is the projected area obtained by the product of the rivet diameter into the plate thickness.

*Associate of Albert Kingsbury, Engineer, Pittsburgh.

In this paper rivet diameter is taken to mean the diameter of the rivet hole.

A joint may fail by the failure of its rivets, or by the failure of the plate by tearing between the rivet holes, or by a combination of the two.

Taking a rivet by itself it is equally resistant to shear and to compression if:*

When in single shear $\frac{\pi d^2 s}{4} = dtc$; i. e. if $\frac{d}{t} = \frac{4c}{\pi s} = k_1$.

When in double shear $\frac{2\pi d^2 S}{4} = dtC$; i. e. if $\frac{d}{t} = \frac{2C}{\pi S} = k_2$

Hence single-shear rivets will fail by shearing if $\frac{d}{t} < k_1$ and by compression if $\frac{d}{t} > k_1$, while double-shear rivets will fail by shear if $\frac{d}{t} < k_2$ and by compression if $\frac{d}{t} > k_2$.

The above is the basis of the whole analysis which is given in detail in the appendix. The results of the analysis as they apply to boiler joint design are condensed into tabular form for convenient reference.

Tables. In one set of tables, II to VI, for general use, values for the unit resistances are not inserted. Hence anyone may use these as a basis, and by inserting the values he is accustomed to use, may easily compile a working set of tables that will yield all the advantages possible in the way of time saving. Another set of tables, VII to XI, is made up from the previous ones, using values for the unit resistances as noted in the tables. These are chosen from the Report of the Boiler Code Committee of the American Society of Mechanical Engineers.

An example is worked on page 285 showing how the tables may be used.

Curves. The analysis has led to the design of a series of curves, Fig. 6–10, showing the joint properties with surprising clearness. They are probably the most interesting feature of the paper because they so clearly interpret the various cases given in the tables. They show at a glance the efficiency of the joint for

*For list of notations see Table I.

a given plate thickness, rivet diameter and pitch; the way the joint will fail; just how it may be improved; the highest possible efficiency and the proper pitches and rivets to use to obtain it; *and how the efficiency varies as the pitch is either increased or decreased from the best pitch for a given rivet diameter.*

NOTATIONS TABLE I

F, G, H, I, J, K, L, M AND N, REPRESENT DIFFERENT MODES OF FAILURE FOR DIFFERENT JOINTS, AND RESISTANCES TO FAILURE IN THESE DIFFERENT WAYS

$a = \frac{\pi d^2}{4}$ = SECTIONAL AREA OF RIVET HOLE ASSUMING THAT THE RIVET FILLS THE HOLE AFTER BEING DRIVEN

b = THICKNESS OF OUTER BUTT-STRAP IN A BUTT-JOINT

c = UNIT RESISTANCE TO CRUSHING BEFORE A RIVET THAT IS SUBJECT TO SINGLE SHEAR

C = " " " " " " " " " " DOUBLE SHEAR. IN THIS DISCUSSION $2a$ IS THE SHEARING AREA OF SUCH A RIVET

d = DIAMETER OF RIVET HOLE SEE a

E = EFFICIENCY OF THE JOINT

f = THICKNESS OF INNER BUTT-STRAP IN A BUTT-JOINT

$k_1 = \frac{4c}{\pi S}$, $k_2 = \frac{2C}{\pi S}$, $k_3 = \frac{4T}{\pi S}$, $k_4 = \frac{T}{C}$. THESE ARE CONSTANTS DEPENDING ON THE UNIT RESISTANCES TO TEARING, SHEARING AND CRUSHING

n = NUMBER OF ROWS OF RIVETS HAVING SAME PITCH AND SUBJECTED TO THE SAME CONDITIONS OF STRESS; e.g. FOR A DOUBLE RIVETED LAP JOINT $n = 2$. IN BUTT-JOINTS ONLY ONE SIDE OF THE CENTER LINE OF THE JOINT IS CONSIDERED

P = PITCH OF RIVETS IN THE OUTER ROW i.e. WHERE THE PITCH IS GREATEST FOR JOINTS HAVING DIFFERENT PITCHES IN DIFFERENT ROWS

s = UNIT RESISTANCE TO SHEARING OF A RIVET THAT IS SUBJECT TO SINGLE SHEAR

S = " " " " " " " " " " DOUBLE SHEAR. IN THIS DISCUSSION $2a$ IS THE SHEARING AREA OF SUCH A RIVET

t = THICKNESS OF THE MAIN PLATES WHICH ARE TO BE JOINTED IF THEY ARE OF UNEQUAL THICKNESS t REPRESENTS THE THINNER

T = UNIT RESISTANCE OF THE PLATE TO TEARING

AREAS = a FOR DIAMETERS = d

d	d	DECIMAL EQUIV.	a
1/4		.250	.0491
	9/32	.2812	.0621
5/16		.3125	.0767
	11/32	.3437	.0928
3/8		.375	.1105
	13/32	.4062	.1296
7/16		.4375	.1503
	15/32	.4687	.1726
1/2		.500	.1964
	9/16	.5625	.2485
5/8		.625	3068
	11/16	.6875	.3712
3/4		.750	.4418
	13/16	.8125	.5185
7/8		.875	.6013
	15/16	.9375	6903
1		1000	.7854
	1 1/16	1.0625	.8866
1 1/8		1.125	.994
	1 3/16	1.1875	1.108
1 1/4		1.250	1.227
	1 5/16	1.3125	1.353
1 3/8		1.375	1.485
	1 7/16	1.4375	1.623
1 1/2		1.500	1.767
	1 9/16	1.5625	1.918
1 5/8		1.625	2.074

TABLE II

LAP-JOINTS *

PITCHES THE SAME IN ALL ROWS.

n = NUMBER OF ROWS.

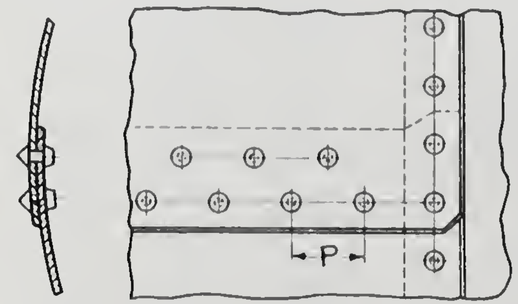
POSSIBLE MODES OF FAILURE AND THE RESISTANCES TO THE.A OFFERED BY THE JOINT FOR A WIDTH EQUAL TO P.

$F = PtT$ = RESISTANCE OFFERED BY A STRIP OF UNPERFORATED PLATE

$G = asn$ = " OF RIVETS TO SHEAR

$H = dtcn$ = " OF RIVETS OR PLATE TO COMPRESSION

$I = (P-d)tT$ = " OF PLATE TO TEARING BETWEEN RIVET HOLES AT OUTER ROW



DOUBLE-RIVETED LAP-JOINT SHOWN

$n = 2$

FIG 1

$$k_1 = \frac{4c}{\pi s}$$

TABLE OF JOINT CHARACTERISTICS

TO USE TABLE CALCULATE FIRST $\frac{d}{t}$

		CASE I	CASE II
CONDITIONS FOR THE CASE		$\frac{d}{t} < k_1$	$\frac{d}{t} > k_1$
POSSIBLE MODES OF FAILURE FOR THE CASE		G OR I	H OR I
BEST PITCH FOR THE CASE		$P_1 = \frac{nas}{tT} + d$	$P_2 = d\left(\frac{nc}{T} + 1\right)$
EFFICIENCY FOR THE BEST PITCH		$E = \frac{G}{F}$ OR $\frac{I}{F}$	$E = \frac{H}{F}$ OR $\frac{I}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	I	I
	EFFICIENCY	$E = \frac{I}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	G	H
	EFFICIENCY	$E = \frac{G}{F} = \frac{nas}{PtT}$	$E = \frac{H}{F} = \frac{ndc}{PT}$

* NOTATIONS AS GIVEN IN TABLE I

TABLE III

BUTT-JOINTS WITH BUTT-STRAPS OF EQUAL WIDTH * PITCHES THE SAME IN ALL ROWS.

n = NUMBER OF ROWS OF RIVETS EITHER SIDE OF CENTER LINE OF JOINT

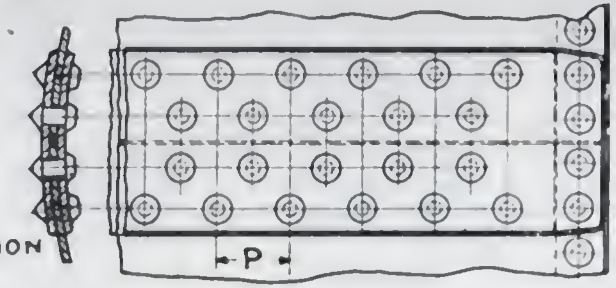
POSSIBLE MODES OF FAILURE AND THE RESISTANCES TO THEM OFFERED BY THE JOINT FOR A WIDTH EQUAL TO P.

$F = PtT$ = RESISTANCE OFFERED BY A STRIP OF UNPERFORATED PLATE.

$G = 2naS$ = " OF RIVETS TO SHEAR

$H = ndtC$ = " OF RIVETS OR PLATE TO COMPRESSION

$I = (P-d)tT$ = " OF PLATE TO TEARING BETWEEN RIVET HOLES AT OUTER ROW



DOUBLE-RIVETED BUTT-JOINT SHOWN $n = 2$

FIG. 2

$$k_2 = \frac{2C}{\pi S}$$

TABLE OF JOINT CHARACTERISTICS

TO USE TABLE, CALCULATE FIRST $\frac{d}{t}$

		CASE I	CASE II
CONDITIONS FOR THE CASE		$\frac{d}{t} < k_2$	$\frac{d}{t} > k_2$
POSSIBLE MODES OF FAILURE FOR THE CASE		G OR I	H OR I
BEST PITCH FOR THE CASE		$P_1 = \frac{2naS}{tT} + d$	$P_2 = d\left(\frac{nC}{T} + 1\right)$
EFFICIENCY F THE BEST PITCH		$E = \frac{G}{F}$ OR $\frac{I}{F}$	$E = \frac{H}{F}$ OR $\frac{I}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	I	I
	EFFICIENCY	$E = \frac{I}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	G	H
	EFFICIENCY	$E = \frac{G}{F} = \frac{2naS}{PtT}$	$E = \frac{H}{F} = \frac{ndC}{PT}$

* NOTATIONS AS GIVEN IN TABLE I

TABLE IV

DOUBLE-RIVETED BUTT-JOINT *
WITH BUTT-STRAPS OF UNEQUAL WIDTH †

POSSIBLE MODES OF FAILURE AND THE RESISTANCES TO THEM OFFERED BY THE JOINT FOR A WIDTH EQUAL TO P.

$F = PtT$ = RESISTANCE OFFERED BY A STRIP OF UNPERFORATED PLATE

$G = (P-d)tT$ = " TO TEARING AT ROW A

$H = (P-2d)tT + as$ = " TO TEARING AT ROW B AND SHEARING RIVETS IN ROW A.

$I = (P-2d)tT + dfc$ = " TO TEARING AT ROW B AND CRUSHING BEFORE RIVETS IN ROW A.

$J = as + 4aS$ = " TO SHEARING ALL THE RIVETS IN ROWS A AND B.

$K = dfc + 2dtC$ = " TO CRUSHING BEFORE ALL THE RIVETS IN ROWS A AND B.

$L = as + 2dtC$ = " TO SHEARING ROW A AND CRUSHING BEFORE RIVETS IN ROW B.

M = SEE DISCUSSION UNDER FIG. 8

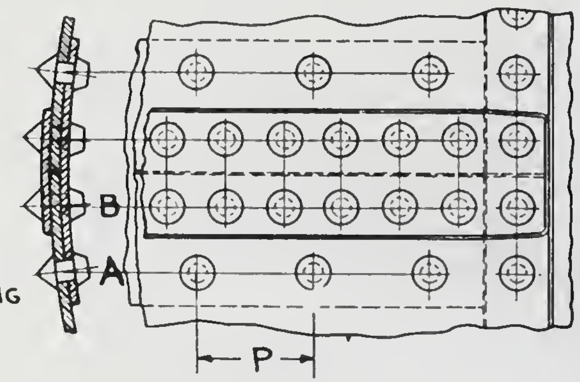


FIG. 3

$K_1 = \frac{4c}{\pi s}$ $K_3 = \frac{4T}{\pi s}$

$K_2 = \frac{2C}{\pi S}$ $K_4 = \frac{T}{C}$

TABLE OF JOINT CHARACTERISTICS TO USE TABLE CALCULATE FIRST $\frac{d}{t}$, $\frac{d}{f}$ AND $\frac{f}{t}$

		CASE I	CASE II (a)	CASE II (b)	CASE III (a)	CASE III (b)
CONDITIONS FOR THE CASE		$\frac{d}{t} < K_2$, $< K_3$; $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $< K_3$; $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $> K_3$; $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $\frac{d}{f} > K_1$, $\frac{f}{t} < K_4$	$\frac{d}{t} > K_2$, $\frac{d}{f} > K_1$, $\frac{f}{t} > K_4$
POSSIBLE MODES OF FAILURE FOR THE CASE		H OR J	H OR L	G OR L	I OR K	G OR K
BEST PITCH FOR THE CASE.		$P_1 = 2d + \frac{4aS}{tT}$	$P_2 = 2d(1 + \frac{C}{T})$	$P_3 = \frac{as}{tT} + d(1 + \frac{2C}{T})$	$P_4 = 2d(1 + \frac{C}{T})$	$P_5 = d(1 + \frac{2C}{T} + \frac{fc}{tT})$
EFFICIENCY FOR THE BEST PITCH.		$E = \frac{H}{F}$ OR $\frac{J}{F}$	$E = \frac{H}{F}$ OR $\frac{L}{F}$	$E = \frac{G}{F}$ OR $\frac{L}{F}$	$E = \frac{I}{F}$ OR $\frac{K}{F}$	$E = \frac{G}{F}$ OR $\frac{K}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	H	H	G	I	G
	EFFICIENCY	$E = \frac{H}{F}$	$E = \frac{H}{F}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	J	L	L	K	K
	EFFICIENCY	$E = \frac{J}{F}$	$E = \frac{L}{F}$	$E = \frac{L}{F}$	$E = \frac{K}{F}$	$E = \frac{K}{F}$

* NOTATIONS AS GIVEN IN TABLE I

† IF $f \geq t$ SUBSTITUTE t FOR f IN RESISTANCES TO FAILURE I AND K AND IN RATIOS d/f AND f/t

TABLE V

TRIPLE-RIVETED BUTT-JOINT * WITH UNEQUAL BUTT-STRAPS †

POSSIBLE MODES OF FAILURE AND THE RESISTANCES TO THEM OFFERED BY THE JOINT FOR A WIDTH EQUAL TO P

$F = PtT$ = RESISTANCE OFFERED BY A STRIP OF UNPERFORATED PLATE

$G = (P-d)tT$ " TO TEARING AT ROW A

$H = (P-2d)tT + as$ " TO TEARING AT ROW B AND SHEARING RIVETS IN ROW A

$I = (P-2d)tT + dfc$ " " TO TEARING AT ROW B AND CRUSHING BEFORE RIVETS IN ROW A

$J = as + 8aS$ " " TO SHEARING ALL THE RIVETS IN ROWS A, B AND C.

$K = dfc + 4dtC$ " " TO CRUSHING BEFORE ALL THE RIVETS IN ROWS A, B AND C

$L = as + 4dtC$ " " TO SHEARING ROW A AND CRUSHING BEFORE RIVETS IN ROWS B AND C

M = SEE DISCUSSION UNDER FIG 9 & 8

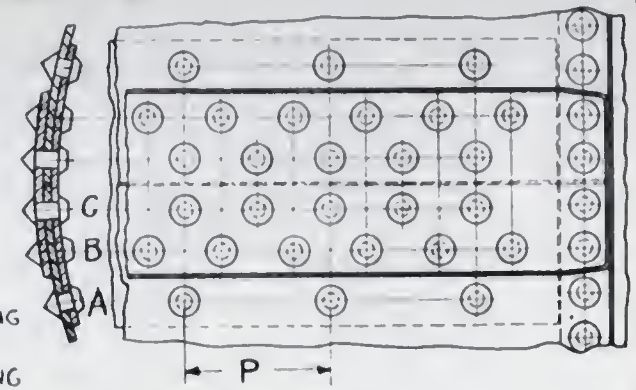


FIG. 4

$K_1 = \frac{4c}{\pi S}$ $K_3 = \frac{4T}{\pi S}$

$K_2 = \frac{2C}{\pi S}$ $K_4 = \frac{T}{C}$

TABLE OF JOINT CHARACTERISTICS. TO USE TABLE CALCULATE FIRST $\frac{d}{t}$, $\frac{d}{f}$ AND $\frac{f}{t}$

		CASE I	CASE II (a)	CASE II (b)	CASE III (a)	CASE III (b)
CONDITIONS FOR THE CASE.		$\frac{d}{t} < K_2$, $< K_3$; $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $< K_3$; $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $> K_3$; $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$; $\frac{d}{f} > K_1$; $\frac{f}{t} < K_4$	$\frac{d}{t} > K_2$; $\frac{d}{f} > K_1$; $\frac{f}{t} > K_4$
POSSIBLE MODES OF FAILURE FOR THE CASE		H OR J	H OR L	G OR L	I OR K	G OR K
BEST PITCH FOR THE CASE		$P_1 = 2d + \frac{8aS}{tT}$	$P_2 = 2d\left(1 + \frac{2C}{T}\right)$	$P_3 = \frac{as}{tT} + d\left(1 + \frac{4C}{T}\right)$	$P_4 = 2d\left(1 + \frac{2C}{T}\right)$	$P_5 = d\left(1 + \frac{4C}{T} + \frac{fc}{tT}\right)$
EFFICIENCY FOR THE BEST PITCH		$E = \frac{H}{F}$ OR $\frac{J}{F}$	$E = \frac{H}{F}$ OR $\frac{L}{F}$	$E = \frac{G}{F}$ OR $\frac{L}{F}$	$E = \frac{I}{F}$ OR $\frac{K}{F}$	$E = \frac{G}{F}$ OR $\frac{K}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	H	H	G	I	G
	EFFICIENCY	$E = \frac{H}{F}$	$E = \frac{H}{F}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	J	L	L	K	K
	EFFICIENCY	$E = \frac{J}{F}$	$E = \frac{L}{F}$	$E = \frac{L}{F}$	$E = \frac{K}{F}$	$E = \frac{K}{F}$

* NOTATIONS AS GIVEN IN TABLE I

† IF $f \geq t$ SUBSTITUTE t FOR f IN RESISTANCES TO FAILURE I AND K AND IN RATIOS d/f AND f/t

TABLE VI

QUADRUPLE-RIVETED BUTT-JOINT *
WITH BUTT-STRAPS OF UNEQUAL WIDTH †

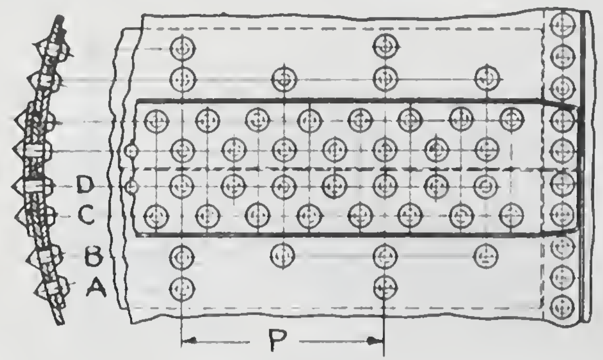


FIG. 5

POSSIBLE MODES OF FAILURE AND THE RESISTANCES TO THEM OFFERED BY THE JOINT FOR A WIDTH EQUAL TO P.

$F = PtT$ = RESISTANCE OFFERED BY A STRIP OF UNPERFORATED PLATE

$G = (P-d)tT$ = " TO TEARING AT ROW A.

$H = (P-2d)tT + aS$ = " TO TEARING AT ROW B AND SHEARING RIVETS IN ROW A

$I = (P-2d)tT + dfC$ = " TO TEARING AT ROW B AND CRUSHING BEFORE RIVETS IN ROW A

$J = (P-4d)tT + 3aS$ = " TO TEARING AT ROW C AND SHEARING AT ROWS A AND B

$K = (P-4d)tT + 3dfC$ = " TO TEARING AT ROW C AND CRUSHING AT ROWS A AND B.

$L = 3aS + 16aS$ = " TO SHEARING ALL RIVETS IN ROWS A, B, C AND D

$M = 3dfC + 8dtC$ = " TO CRUSHING BEFORE ALL RIVETS IN ROWS A, B, C AND D.

$N = 3aS + 8dtC$ = " TO SHEARING ROWS A AND B AND CRUSHING BEFORE ROWS C AND D

Q = SEE DISCUSSION UNDER FIG. 10

$K_1 = \frac{4C}{\pi S}$ $\quad K_3 = \frac{4T}{\pi S}$.

$K_2 = \frac{2C}{\pi S}$ $\quad K_4 = \frac{T}{C}$

TABLE OF JOINT CHARACTERISTICS. TO USE TABLE CALCULATE FIRST $\frac{d}{t}$, $\frac{d}{f}$ AND $\frac{f}{t}$

		CASE I	CASE II (a)	CASE II (b)	CASE III (a)	CASE III (b)
CONDITIONS FOR THE CASE		$\frac{d}{t} < K_2$, $\frac{f}{t} < K_3$, $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $\frac{f}{t} < K_3$, $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $\frac{f}{t} > K_3$, $\frac{d}{f} < K_1$	$\frac{d}{t} > K_2$, $\frac{d}{f} > K_1$, $\frac{f}{t} < K_4$	$\frac{d}{t} > K_2$, $\frac{d}{f} > K_1$, $\frac{f}{t} > K_4$
POSSIBLE MODES OF FAILURE FOR THE CASE		J OR L	J OR N	G OR N	K OR M	G OR M
BEST PITCH FOR THE CASE		$P_1 = 4d + \frac{16aS}{tT}$	$P_2 = 4d\left(1 + \frac{2C}{T}\right)$	$P_3 = \frac{3aS}{tT} + d\left(1 + \frac{8C}{T}\right)$	$P_4 = 4d\left(1 + \frac{2C}{T}\right)$	$P_5 = d\left(1 + \frac{8C}{T} + \frac{3fC}{tT}\right)$
EFFICIENCY FOR THE BEST PITCH.		$E = \frac{J}{F}$ OR $\frac{L}{F}$	$E = \frac{J}{F}$ OR $\frac{N}{F}$	$E = \frac{G}{F}$ OR $\frac{N}{F}$	$E = \frac{K}{F}$ OR $\frac{M}{F}$	$E = \frac{G}{F}$ OR $\frac{M}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	J	J	G	K	G
	EFFICIENCY	$E = \frac{J}{F}$	$E = \frac{J}{F}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$	$E = \frac{K}{F}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	L	N.	N	M	M
	EFFICIENCY	$E = \frac{L}{F}$	$E = \frac{N}{F}$	$E = \frac{N}{F}$	$E = \frac{M}{F}$	$E = \frac{M}{F}$

* NOTATIONS AS GIVEN IN TABLE I

† IF $f \geq t$ SUBSTITUTE t FOR f IN RESISTANCES TO FAILURE I AND K AND IN RATIOS d/f AND f/t

TABLE VII

LAP-JOINTS *

PITCHES THE SAME IN ALL ROWS

n = NUMBER OF ROWS

$T = 55{,}000$ $c = 95{,}000$
$S = 44{,}000$ $K_1 = 2.75$

		CASE I	CASE II
CONDITIONS FOR THE CASE		$\frac{d}{t} < 2.75$	$\frac{d}{t} > 2.75$
POSSIBLE MODES OF FAILURE FOR THE CASE.		G OR I	H OR I
BEST PITCH FOR THE CASE.		$P_1 = d + \frac{.8na}{t}$	$P_2 = d(1 + 1.727n)$
EFFICIENCY FOR THE BEST PITCH		$E = \frac{G}{F}$ OR $\frac{I}{F}$	$E = \frac{H}{F}$ OR $\frac{I}{F}$
FOR LESSER PITCHES	MODE OF FAILURE.	I	I
	EFFICIENCY	$E = \frac{I}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	G	H
	EFFICIENCY	$E = \frac{G}{F} = \frac{.8an}{Pt}$	$E = \frac{H}{F} = \frac{1.727dn}{P}$

TABLE VIII

BUTT-JOINTS WITH STRAPS OF EQUAL WIDTH *

PITCHES THE SAME IN ALL ROWS

n - NUMBER OF ROWS OF RIVETS EITHER SIDE OF CENTER LINE OF JOINT

$T = 55{,}000$ $C = 95{,}000$
$S = 44{,}000$ $K_2 = 1.375$

		CASE I	CASE II
CONDITIONS FOR THE CASE		$\frac{d}{t} < 1.375$	$\frac{d}{t} > 1.375$
POSSIBLE MODES OF FAILURE FOR THE CASE		G OR I	H OR I
BEST PITCH FOR THE CASE.		$P_1 = d + \frac{1.6na}{t}$	$P_2 = d(1 + 1.727n)$
EFFICIENCY FOR THE BEST PITCH		$E = \frac{G}{F}$ OR $\frac{I}{F}$	$E = \frac{H}{F}$ OR $\frac{I}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	I	I
	EFFICIENCY	$E = \frac{I}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	G	H
	EFFICIENCY	$E = \frac{G}{F} = \frac{1.6an}{Pt}$	$E = \frac{H}{F} = \frac{1.727dn}{P}$

TABLE IX

DOUBLE-RIVETED BUTT-JOINT *

WITH BUTT-STRAPS OF UNEQUAL WIDTH †

$T = 55{,}000$ $c = C = 95{,}000$ $K_1 = 2.75$ $K_3 = 1.592$
$s = S = 44{,}000$ $K_2 = 1.375$ $K_4 = .579$

		CASE I	CASE II(a)	CASE II(b)	CASE III(a)	CASE III(b)
CONDITIONS FOR THE CASE		$\frac{d}{t} < 1.375$ $\frac{d}{f} < 2.75$	$\frac{d}{t}\begin{cases} > 1.375 \\ < 1.592 \end{cases}$ $\frac{d}{f} < 2.75$	$\frac{d}{t} > 1.592$ $\frac{d}{f} < 2.75$	$\frac{d}{t} > 1.375$ $\frac{d}{f} > 2.75$ $\frac{f}{t} < .579$	$\frac{d}{t} > 1.375$ $\frac{d}{f} > 2.75$ $\frac{f}{t} > .579$
POSSIBLE MODES OF FAILURE FOR THE CASE		H OR J	H OR L	G OR L	I OR K	G OR K
BEST PITCH FOR THE CASE		$P_1 = 2d + \frac{3.2a}{t}$	$P_2 = 5.454d$	$P_3 = \frac{8a}{t} + 4.45d$	$P_4 = 5.454d$	$P_3 = d(4.45 + \frac{1.727f}{t})$
EFFICIENCY FOR THE BEST PITCH.		$E = \frac{H}{F}$ OR $\frac{J}{F}$	$E = \frac{H}{F}$ OR $\frac{L}{F}$	$E = \frac{G}{F}$ OR $\frac{L}{F}$	$E = \frac{I}{F}$ OR $\frac{K}{F}$	$E = \frac{G}{F}$ OR $\frac{K}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	H	H	G	I	G
	EFFICIENCY	$E = \frac{H}{F} = 1 - \frac{2d}{P} + \frac{8a}{Pt}$	$E = \frac{H}{F} = 1 - \frac{2d}{P} + \frac{8a}{Pt}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$	$E = \frac{I}{F} = 1 - \frac{2d}{P} + \frac{1.727df}{Pt}$	$E = \frac{G}{F} = 1 - \frac{d}{P}$
FOR GREATER PITCHES	MODE OF FAILURE	J	L	L	K	K
	EFFICIENCY	$E = \frac{J}{F} = \frac{4a}{Pt}$	$E = \frac{L}{F} = \frac{8a}{Pt} + \frac{3.45d}{P}$	$E = \frac{L}{F} = \frac{8a}{Pt} + \frac{3.45d}{P}$	$E = \frac{K}{F} = \frac{1.727df}{Pt} + \frac{3.45d}{P}$	$E = \frac{K}{F} = \frac{1.727df}{Pt} + \frac{3.45d}{P}$

*NOTATIONS AS GIVEN IN TABLE I

† IF $f \geqq t$ SUBSTITUTE t FOR f IN RESISTANCES TO FAILURE I AND K AND IN RATIOS d/f AND f/t.

TABLE X

TRIPLE-RIVETED BUTT-JOINT *

WITH UNEQUAL BUTT-STRAPS †

$T = 55{,}000$ $c = C = 95{,}000$ $K_1 = 2.75$ $K_3 = 1.592$

$s = S = 44{,}000$ $K_2 = 1.375$ $K_4 = .579$

		CASE I	CASE II (a)	CASE II (b)	CASE III (a)	CASE III (b)
CONDITIONS FOR THE CASE		$\frac{d}{t} < 1.375$ $\frac{d}{f} < 2.750$	$\frac{d}{t}\begin{cases} >1.375 \\ <1.592 \end{cases}$ $\frac{d}{f} < 2.75$	$\frac{d}{t} > 1.592$ $\frac{d}{f} < 2.750$	$\frac{d}{t} > 1.375$ $\frac{d}{f} > 2.750$ $\frac{f}{t} < .579$	$\frac{d}{t} > 1.375$ $\frac{d}{f} > 2.750$ $\frac{f}{t} > .579$
POSSIBLE MODES OF FAILURE FOR THE CASE		H OR J	H OR L	G OR L	I OR K	G OR K
BEST PITCH FOR THE CASE		$P_1 = 2d + 6.4\frac{a}{t}$	$P_2 = 8.91d$	$P_3 = .8\frac{a}{t} + 7.91d$	$P_4 = 8.91d$	$P_5 = (7.91 + 1.727\frac{f}{t})d$
EFFICIENCY FOR THE BEST PITCH		$E = \frac{H}{F}$ OR $\frac{J}{F}$	$E = \frac{H}{F}$ OR $\frac{L}{F}$	$E = \frac{G}{F}$ OR $\frac{L}{F}$	$E = \frac{I}{F}$ OR $\frac{K}{F}$	$E = \frac{G}{F}$ OR $\frac{K}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	H	H	G	I	G
	EFFICIENCY	$E = \frac{H}{F} = 1 - \frac{2d}{p} + \frac{.8a}{pt}$	$E = \frac{H}{F} = 1 - \frac{2d}{p} + \frac{.8a}{pt}$	$E = \frac{G}{F} = 1 - \frac{d}{p}$	$E = \frac{I}{F} = 1 - \frac{2d}{p} + 1.727\frac{df}{pt}$	$E = \frac{G}{F} = 1 - \frac{d}{p}$
FOR GREATER PITCHES	MODE OF FAILURE	J	L	L	K	K
	EFFICIENCY	$E = \frac{J}{F} = \frac{7.2a}{pt}$	$E = \frac{L}{F} = \frac{.8a}{pt} + \frac{6.91d}{p}$	$E = \frac{L}{F} = \frac{.8a}{pt} + \frac{6.91d}{p}$	$E = \frac{K}{F} = \frac{1.727df}{pt} + \frac{6.91d}{p}$	$E = \frac{K}{F} = \frac{1.727df}{pt} + \frac{6.91d}{p}$

TABLE XI

QUADRUPLE-RIVETED BUTT-JOINT *

WITH BUTT-STRAPS OF UNEQUAL WIDTH †

$T = 55{,}000$ $K_1 = 2.75$ $K_3 = 1.592$

$s = S = 44{,}000$ $c = C = 95{,}000$ $K_2 = 1.375$ $K_4 = .579$

		CASE I	CASE II (a)	CASE II (b)	CASE III (a)	CASE III (b)
CONDITIONS FOR THE CASE		$\frac{d}{t} < 1.375$ $\frac{d}{t} < 2.75$	$\frac{d}{t}\begin{cases} >1.375 \\ <1.592 \end{cases}$ $\frac{d}{f} < 2.75$	$\frac{d}{t} > 1.59$ $\frac{d}{f} < 2.75$	$\frac{d}{t} > 1.375$ $\frac{d}{f} < 2.75$ $\frac{f}{t} < .579$	$\frac{d}{t} > 1.375$ $\frac{d}{f} > 2.75$ $\frac{f}{t} > .579$
POSSIBLE MODES OF FAILURE FOR THE CASE		J OR L	J OR N	G OR N	K OR M	G OR M
BEST PITCH FOR THE CASE		$P_1 = 4d + 12.8\frac{a}{t}$	$P_2 = 17.82d$	$P_3 = 2.4\frac{a}{t} + 14.82d$	$P_4 = 17.82d$	$P_5 = d(14.82 + 5.18\frac{f}{t})$
EFFICIENCY FOR THE BEST PITCH		$E = \frac{J}{F}$ OR $\frac{L}{F}$	$E = \frac{J}{F}$ OR $\frac{N}{F}$	$E = \frac{G}{F} + \frac{N}{F}$	$E = \frac{K}{F}$ OR $\frac{M}{F}$	$E = \frac{G}{F}$ OR $\frac{M}{F}$
FOR LESSER PITCHES	MODE OF FAILURE	J	J	G	K	G
	EFFICIENCY	$E = \frac{J}{F} = 1 - \frac{4d}{p} + \frac{2.4a}{pt}$	$E = \frac{J}{F} = 1 - \frac{4d}{p} + \frac{2.4a}{pt}$	$E = \frac{G}{F} = 1 - \frac{d}{p}$	$E = \frac{K}{F} = 1 - \frac{4d}{p} + \frac{5.18fd}{pt}$	$E = \frac{G}{F} = 1 - \frac{d}{p}$
FOR GREATER PITCHES	MODE OF FAILURE	L	N	N	M	M
	EFFICIENCY	$E = \frac{L}{F} = \frac{15.2a}{pt}$	$E = \frac{N}{F} = \frac{2.4a}{pt} + \frac{13.82d}{p}$	$E = \frac{N}{F} = \frac{2.4a}{pt} + \frac{13.82d}{p}$	$E = \frac{M}{F} = \frac{5.18df}{pt} + \frac{13.82d}{p}$	$E = \frac{M}{F} = \frac{5.18df}{pt} + \frac{13.82d}{p}$

* NOTATIONS AS GIVEN IN TABLE I

† IF $f \leqq t$ SUBSTITUTE t FOR f IN RESISTANCES TO FAILURE I AND K AND IN RATIOS $\frac{d}{f}$ AND $\frac{f}{t}$

One method of using the tables is illustrated by the following:

Example—Find the efficiency of a triple-riveted butt-joint with butt-straps of unequal width, using the following proportions:

Thickness of plate = t = ⅜.

Thickness of inner strap = f = 5/16.

Pitch in outer row = p = 6½.

Diameter of rivet hole = d = 13/16.

First we refer to Table X which shows the properties of this joint. Then we find that $\frac{d}{t} = 2.167$, which is greater than k_3 or 1.592. Next we find that $\frac{d}{f} = 2.6$, which is less than k_1 or 2.75. The values of these ratios place our problem in case II(b).

The modes of failure possible in this case are G and L. Referring back to Table IV we may calculate the resistances to failure from the formulas there given. We find the following:

G = 117 000; L = 138 500; F = 134 000.

Hence the joint will fail by mode G—that is, by tearing at the outer row of rivets. The efficiency of the joint will be $\frac{G}{F} = \frac{117\,300}{134\,000} = 0.875$.

The above example is the same one as given in the Report of the Boiler Code Committee published in the *Transactions* of the American Society of Mechanical Engineers. The results check. You will note, however, that in solving the problem, we did not figure resistances to failure H, I, J, and K, but instead calculated the simple ratios $\frac{d}{t}$ and $\frac{d}{f}$.

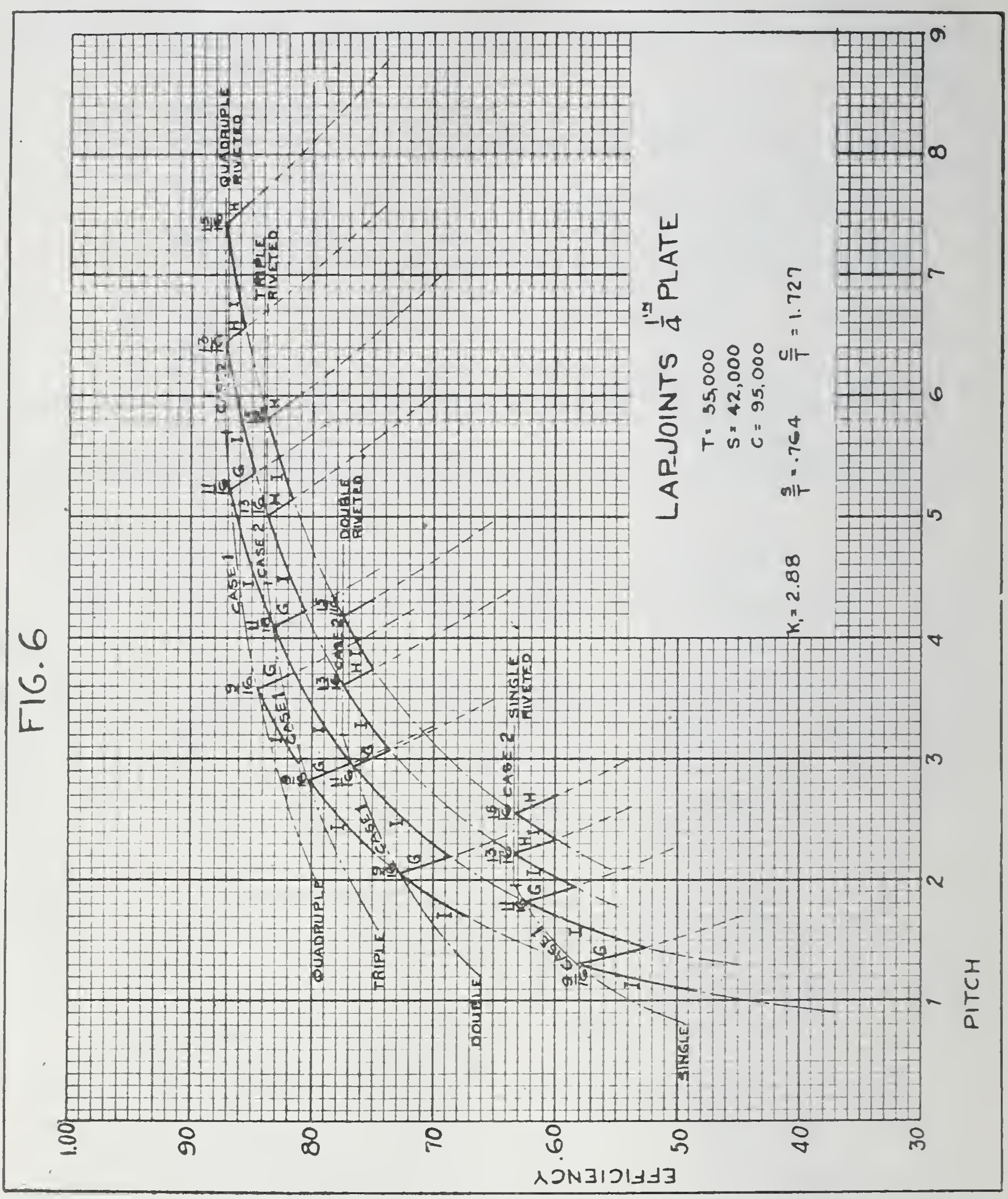

LAP-JOINTS 1/4" PLATE
T = 55,000
S = 42,000
C = 95,000
K = 2.88
S/T = .764
C/T = 1.727
QUADRUPLE RIVETED
TRIPLE RIVETED
DOUBLE RIVETED
SINGLE RIVETED
QUADRUPLE
TRIPLE
DOUBLE
SINGLE
CASE 1
CASE 2
EFFICIENCY
PITCH

FIG. 6

Fig. 6 shows a series of curves calculated for a ¼-inch plate, using the unit resistances noted in the figure. We have drawn the curves for four lap-joints—single-, double-, triple-, and quadruple-riveted. We have shown the best pitch and efficiency for four rivets for each type of joint.

Each inverted-V pair of curves shows the relation between pitch and efficiency for the rivet-hole diameter marked at the apex, and for the plate thickness noted in the figure.

Take, for example, a single-riveted joint. The best pitch for a 9/16 rivet hole is 1.3 inch and the efficiency is 0.58. If the pitch is increased the efficiency falls off rapidly along the curve marked G. G also represents the mode of failure indicated by that letter in Table II. If the pitch is decreased the efficiency falls off along the line I. I also represents the mode of failure.

The dot and dash line connecting the apexes of the curves, for each type of joint shown in this figure, undergoes a marked change of direction between the 11/16 and 13/16 rivet holes. To the left of this point the rivets fall in case I, and to the right in case II. It is evident that, if joint efficiency is the only consideration, we should always use rivet holes that lie in case II. All rivet falling in case II will give the maximum efficiency that can be obtained for this joint; but the larger the rivet the greater should be the pitch.

Let us assume that the pitch should be two inches, or eight times the plate thickness, in order to calk well. We see from the curves that the 11/16-inch hole gives us 0.57 efficiency, while the 13/16-inch gives nearly 0.60. If we begin by assuming a rivet hole diameter, 13/16 for example, we find the most efficient pitch to be 2.23 inches. We may have to depart from this pitch slightly in order to get one that will divide into the seam length without remainder, or one that will be standard and that will calk properly. The curves show at once the influence of such departure upon the joint efficiency.

Comparison between the single-, double-, triple- and quadruple-riveted lap-joints brings out several interesting features:

1. The possible efficiency increases with the number of rows, rapidly at first.

2. For a given rivet diameter the most efficient pitch is greater as the number of rows is increased.

3. Departure from the most efficient pitch has a smaller relative inuence on the efficiency as the number of rows increases.

4. For a given pitch the most efficient rivet diameter decreases as the number of rows increases.

5. As the number of rows is increased the joint efficiency is affected relatively less by reducing than by increasing the pitch from that which is best for a given rivet.

It will be noted that the curves in Fig. 6 are drawn for a given plate thickness and certain definite rivet hole diameters and pitches. They might have been drawn so as to apply to all plate thicknesses and rivets provided $\frac{d}{t}$ had been substituted for d, and $\frac{p}{t}$ for p. This remark applies equally to Fig. 7, and with some limitations to Fig. 8–10.

It occurs to the author that curves of this kind could be used for the interpretation of test data on the strength of riveted joints so as to determine whether the constants for tension, shear, and compression, as now used, have been wisely chosen.

Fig. 7 shows a set of curves for butt-joints with two straps of equal width. These are drawn for ¼-inch plate. The unit resistances to failure are noted in the figure. The interpretation of the curves is the same as explained for Fig. 6. A direct comparison between Fig. 6 and Fig. 7 cannot be made because the figure for shearing resistance is, unfortunately, not quite the same for both sets of curves. It is quite evident, however, that the same pitch requires a smaller rivet in the butt-joint than in the lap-joint to give the maximum efficiency. Another obvious feature is that a lap-joint with n rows of rivets proves fully as efficient as this type of butt-joint with n rows of rivets each side of the center-line.

Fig. 8 shows a set of curves for a double-riveted butt-joint with unequal width straps as drawn in Fig. 3. These are drawn for a ⅜-inch main plate and a 5/16-inch inner butt-strap. The unit resistances are as noted in the figure.

The dot and dash line connecting the apexes of the curves

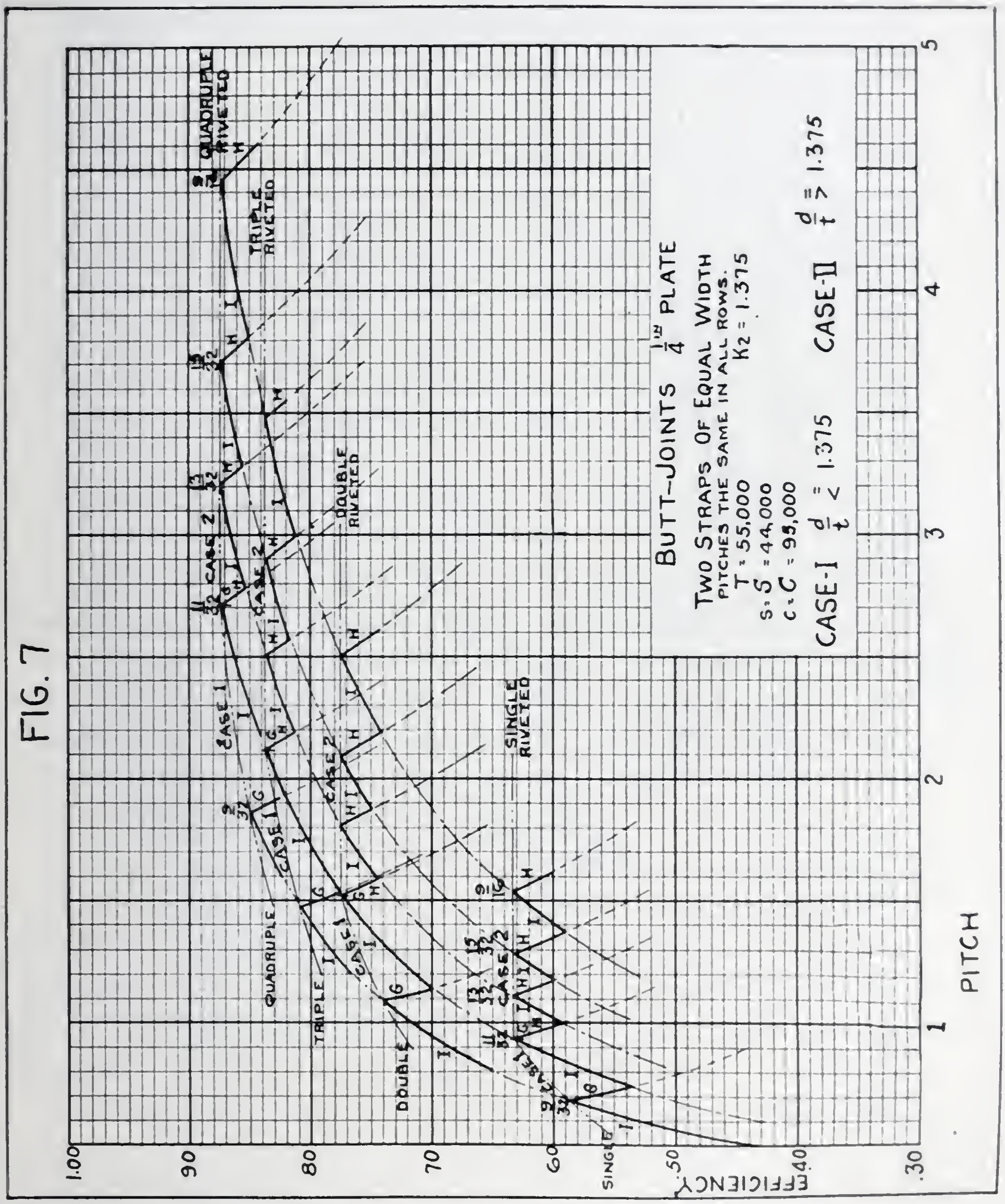

BUTT-JOINTS 1/4" PLATE
TWO STRAPS OF EQUAL WIDTH
PITCHES THE SAME IN ALL ROWS.
T = 55,000
S = 44,000
C = 95,000
K2 = 1.375
CASE-I d/t ≤ 1.375
CASE-II d/t > 1.375
QUADRUPLE RIVETED
TRIPLE RIVETED
DOUBLE RIVETED
SINGLE RIVETED
PITCH
EFFICIENCY

FIG. 7

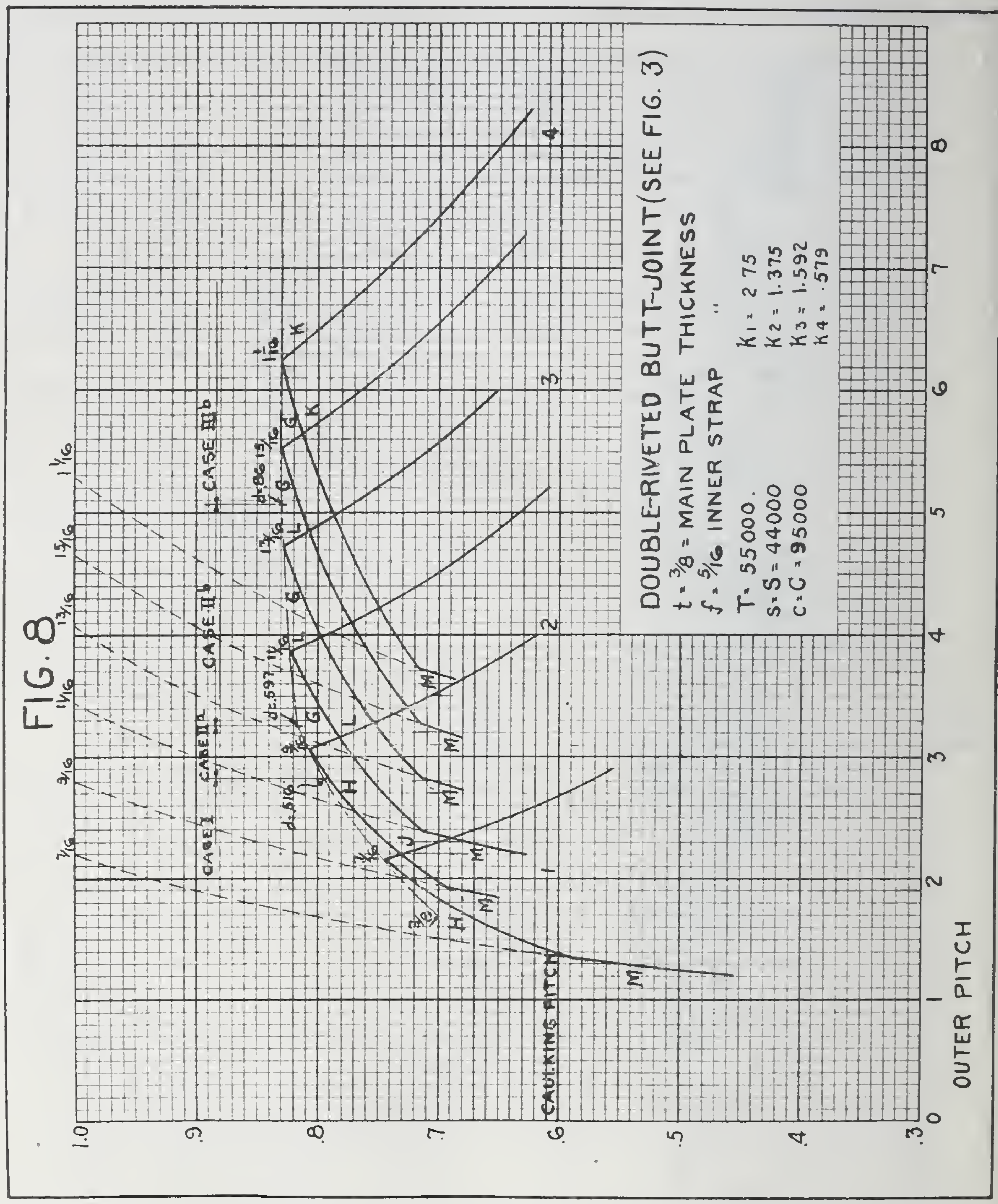

DOUBLE-RIVETED BUTT-JOINT (SEE FIG. 3)
t = 3/8 = MAIN PLATE THICKNESS
f = 5/16 INNER STRAP "
T = 55000.
S = S = 44000
C = C = 95000
K1 = 2.75
K2 = 1.375
K3 = 1.592
K4 = .579
CASE I
CASE II a
CASE II b
CASE III b
CAULKING PITCH
OUTER PITCH

FIG. 8

is marked at intervals to indicate the limits of case I, case II(a), case II(b) and case III(b). The inner butt-strap is too thick for case III(a) to appear. Cases III(a) and III(b) cannot occur at the same time for a given set of plates.

In general the interpretation of these curves is the same as described under Fig. 6. One new feature appears here which should also be borne in mind for Fig. 9–10. It is as follows:

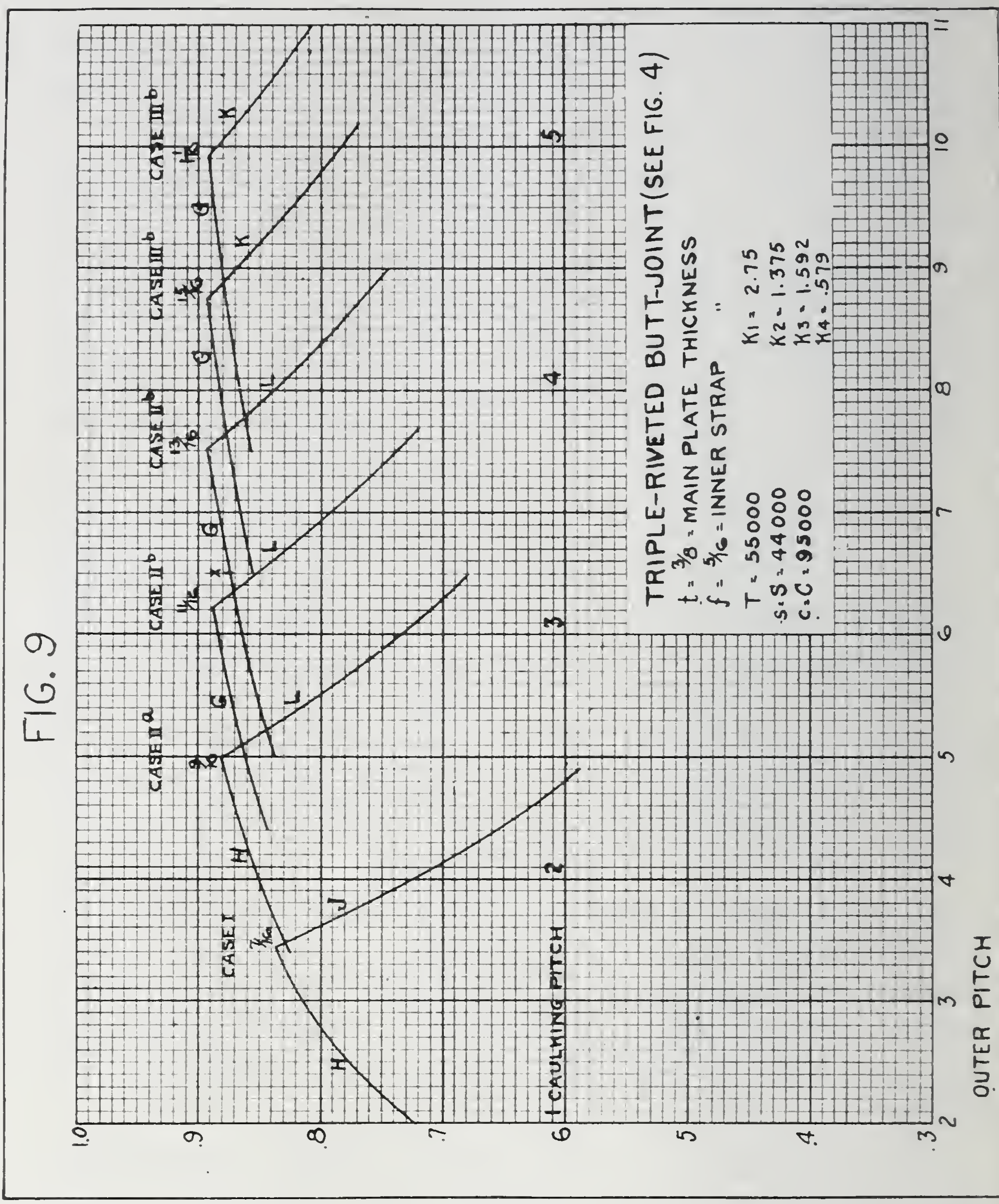
When joints of this type are designed with proportions departing widely from those that show up best in this set of curves it is necessary to take account of the possibility of the failure of the joint by the tearing of the inner and outer butt-straps between the rivets that are in double shear. This mode of failure is more liable to occur when the butt-straps are relatively thin compared with the main plates. For this joint the resistance to this mode of failure will be denoted by $M = (P - 2d)\ (b + f)T$.

In Fig. 8 the set of dotted curves marked 7/16, 9/16, 11/16, 13/16, and 1–1/16, at their upper ends, are drawn by plotting $\frac{M}{F}$ against P for each value of d. When these dotted curves reach the corresponding full-line curves from the several apexes the dotted curves are continued as full lines. This is to indicate that from these points on they must be taken into account. It can be shown that the positions of these points are relatively nearer the apexes of the full-line curves when the butt-straps are relatively thinner compared with the main plates.

For example, consider the 11/16 rivet hole. The most efficient pitch is about 3⅞ in the outer row. If the pitch be reduced the efficiency falls off from 0.82 along the line G till it reaches a value of 0.71 at a pitch of 2⅜. A further reduction of the pitch causes the efficiency to fall off very rapidly along the line M.

It may be stated here that abnormal designs, involving thin butt-straps, or one thick and the other thin, would lead to other possible modes of failure which are not usually taken into account at all. Hence abnormal designs should be avoided when possible.

It will be noted that in Fig. 8 both the outer and inner pitches are shown, but by different scales.



TRIPLE-RIVETED BUTT-JOINT (SEE FIG. 4)
t = 3/8 = MAIN PLATE THICKNESS
f = 5/16 = INNER STRAP "
T = 55000
s.S = 44000
c.C = 95000
K1 = 2.75
K2 = 1.375
K3 = 1.592
K4 = .579
CASE I
CASE II a
CASE II b
CASE III b
CAULKING PITCH
OUTER PITCH

FIG. 9

Fig. 9 shows a set of curves for a triple-riveted butt-joint with two straps of unequal width as drawn in Fig. 4. These curves are drawn for a ⅜-inch main plate and a 5/16-inch inner butt-strap. The unit resistances are as noted in the figure. The remarks made under Fig. 8 would apply in general also to Fig. 9. It should be noted that the left branches of the curves are more nearly horizontal here than in Fig. 8. For this joint the resistance to failure by the tearing of the two butt-straps between the rivets in double shear will be denoted by $M = (P - 2d)(b + f)T$. The M-curves are not shown. See remarks on this subject under Fig. 8.

The solution of the example worked out on page 285 is indicated by the point **x** on the left branch of the curve for 13/16-inch rivet holes.

Fig. 10 shows a set of curves for a quadruple-riveted butt-joint with two straps of unequal width as drawn in Fig. 5. These curves are drawn for a ⅜-inch main plate and a 5/16-inch inner butt-strap. The unit resistances are as noted in the figure. The remarks made under Fig. 8 would apply in general also to Fig. 10.

It should be noted that the left branches of the curves are more nearly horizontal than in either Fig. 8 or Fig. 9. Note also that the curve for a rivet hole in case I lies wholly below the left branch of a curve for a rivet hole in case II(a).

For this joint the resistance to failure by the tearing of the two butt-straps between the rivets in double shear will be denoted by $Q = (P - 4d)(b + f)T$. Only one Q-curve is shown—that for the 9/16-inch rivet hole. It does not have to be taken into account till we reduce the pitch for this rivet from 10 inches to 4⅝ inches in the outer row. Such a small pitch would be very abnormal.

APPENDIX—DETAILED ANALYSIS

LAP-JOINT

Pitches the same in all *n* rows of rivets. (See Fig. 1, Table II, and notations in Table I.)

This joint may fail by method G, H, or I, as indicated in Table II. The object of this analysis is to determine the conditions under which it will fail by one mode or another.

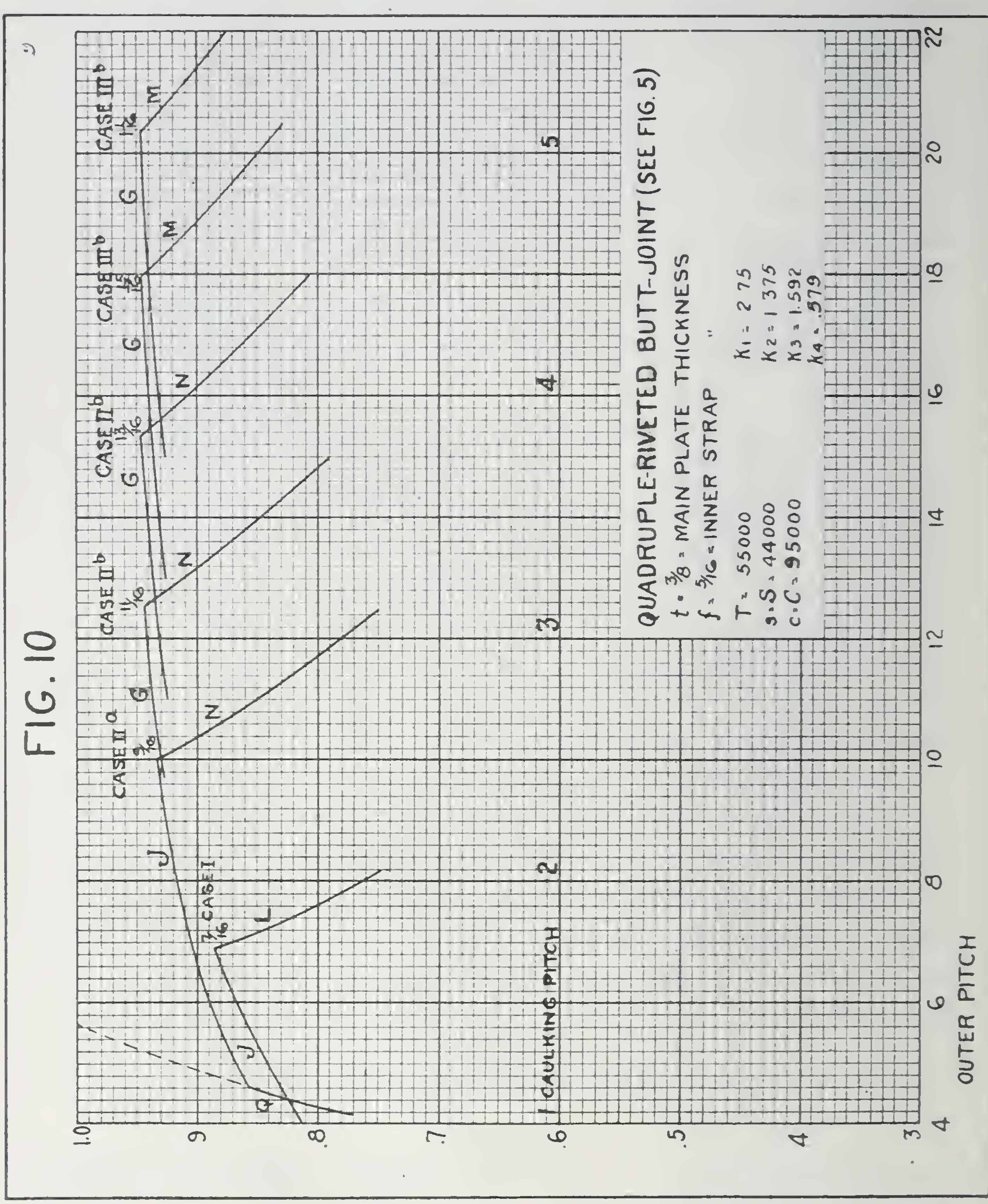

QUADRUPLE-RIVETED BUTT-JOINT (SEE FIG. 5)
t = 3/8 = MAIN PLATE THICKNESS
f = 5/16 = INNER STRAP "
T = 55000
s·S = 44000
c·C = 95000
K1 = 2.75
K2 = 1.375
K3 = 1.592
K4 = .579
CASE I
CASE II a
CASE II b
CASE III b
CAULKING PITCH
OUTER PITCH

FIG. 10

Equate the resistance to shear, G, with that to compression, H. In place of a use $\frac{\pi d^2}{4}$. We get

$\frac{\pi d^2 sn}{4} = dtcn$. Solving this for the ratio $\frac{d}{t}$ we get

$\frac{d}{t} = \frac{4c}{\pi s} = k_1$.

The best* rivet to use for lap-joints will be that which satisfies $\frac{d}{t} = k_1$. If $\frac{d}{t} < k_1$ the rivets are weaker in shear than in compression (call this case I), while if $\frac{d}{t} > k_1$ the rivets are weaker in compression than in shear (call this case II).

Now consider the above cases in more detail.

Case I: $\frac{d}{t} < k_1$. The *rivets* can fail only by shearing. Hence the joint may fail only by mode G or mode I. The best pitch—i. e. the one that gives the maximum efficiency for a given rivet diameter—for this case will occur when G = I, or

$asn = (P - d)tT = PtT - dtT$.

$P = \frac{asn}{tT} + d = P_1$.

For this pitch the joint will be as likely to fail by G as by I. The efficiency would be found by $E = \frac{G}{F}$ or $\frac{I}{F}$. If in case I a pitch less than P_1 be used, the joint will fail by mode I, and we have $E = \frac{I}{F} = 1 - \frac{d}{p}$. If, however, a pitch greater than P_1 be used, the joint will fail by mode G, and we have $E = \frac{G}{F}$.

Case II: $\frac{d}{t} > k_1$. The rivets can fail only by compression; hence the joint may fail only by mode H or mode I. The best pitch for this case will occur when H = I, or

$dtcn = (P - d)tT = PtT - dtT$.

*By best rivet is meant that which will give the highest efficiency with the minimum pitch. All rivets larger than this give the same efficiency, but with increasingly greater pitches.

$$P = d\left(\frac{cn}{T} + 1\right) = P_2.$$

For this pitch the joint will be as likely to fail by H as by I. The efficiency would be found by $E = \frac{H}{F}$ or $\frac{I}{F}$. If in case II, a pitch less than P_2 be used, the joint will fail by mode I, and we have $E = \frac{I}{F} = 1 - \frac{d}{p}$. If, however, a pitch greater than P_2 be used, the joint will fail by mode H, and we have $E = \frac{H}{F}$.

BUTT-JOINT

Butt-straps of equal width. Pitches the same in all *n* rows. (See Fig. 2, Table III, and notations in Table I.)

This joint may fail by method G, H, or I, as indicated in Table III. The object of this analysis is to determine the conditions under which it will fail by one mode or another.

Equate the resistance to shear, G, with that to compression, H. In place of a use $\frac{\pi d^2}{4}$. We get

$\frac{2\pi d^2 nS}{4} = ndtC$, from which we get

$\frac{d}{t} = \frac{2C}{\pi S} = k_2$. The best rivet to use for this type of butt-joint will be that which satisfies $\frac{d}{t} = k_2$, because if $\frac{d}{t} < k_2$ the rivets are weaker in shear than in compression (call this case I), while if $\frac{d}{t} > k_2$ the rivets are weaker in compression than in shear (call this case II).

Considering the above cases in more detail we have

Case I: $\frac{d}{t} < k_2$. The rivets can fail only by shearing. Hence the joint may fail only by mode G or mode I. The best pitch for this case will occur when $G \doteq I$, or

$$2naS = (P - d)tT = PtT - dtT$$

$$P = \frac{2naS}{tT} + d = P_1.$$

For this pitch the joint will be as likely to fail by G as by I.

The efficiency would be found by $E = \frac{G}{F}$ or $\frac{I}{F}$. If in case I a pitch less than P_1 be used, the joint will fail by mode I and the efficiency will be $E = \frac{I}{F} = 1 - \frac{d}{p}$. If, however, a pitch greater than P_1 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F}$.

Case II: $\frac{d}{t} > k_2$. The rivets can fail only by compression. Hence the joint may fail only by mode H or mode I. The best pitch for this case will occur when $H = I$, or

$$ndtC = (P - d)tT = PtT - dtT.$$

$$P = d\left(\frac{nC}{T} + 1\right) = P_2.$$

For this pitch the joint is as likely to fail by H as by I. The efficiency would be found by $E = \frac{H}{F}$ or $\frac{I}{F}$. If in case II, a pitch less than P_2 be used, the joint will fail by mode I and the efficiency will be $E = \frac{I}{F} = 1 - \frac{d}{p}$. If, however, a pitch greater than P_2 be used, the joint will fail by mode H and the efficiency will be $E = \frac{H}{F}$

DOUBLE-RIVETED BUTT-JOINT

Butt-straps of unequal width. (See Fig. 3, Table IV, and notations in Table I.)

This joint may fail in any one of the six ways, G to L inclusive, indicated in Table IV. The object of this analysis is to determine the conditions under which it will fail in one way rather than another. In this joint the outer row contains only rivets which are in *single* shear, while those in the inner row are in *double* shear. We have already determined that a rivet in single shear is equally resistant to shear and to compression when $\frac{d}{t} = k_1$. In this joint we have to take into account the butt-strap thicknesses, b (outer) and f (inner), as shown by Fig. 3. Usually f is less than t; hence we will use f instead of t because a rivet in single shear will crush in front of the thinner of the two plates. Thus

we have $\frac{d}{f} = k_1$.

If $\frac{d}{f} < k_1$, rivets in row A will shear rather than crush.
If $\frac{d}{f} > k_1$, rivets in row A will crush rather than shear.

We have also already determined that a rivet in double shear is equally resistant to shear and to compression when $\frac{d}{t} = k_2$. Here we neglect the butt-strap thicknesses because b + f is assumed to be greater than t, and the failure by compression takes place at the main plate.

If $\frac{d}{t} < k_2$, rivets in row B will shear rather than crush.
If $\frac{d}{t} > k_2$, rivets in row B will crush rather than shear.

Thus we have the following conditions to deal with:

$$\frac{d}{f} < k\,;\ \frac{d}{f} > k\,;\ \frac{d}{t} < k_2\,;\ \frac{d}{t} > k_2.$$

Certain pairs of these conditions may occur at the same time in a joint. As far as this analysis is concerned, f < t, and $k_1 > k_2$. Hence the number of combinations possible reduces to the three following if $\frac{f}{t} \overset{=}{<} \frac{1}{2}\,\frac{Cs}{cS}$.

Case I: $\frac{d}{t} < k_2$, and $\frac{d}{f} < k_1$.

Case II: $\frac{d}{t} > k_2$, and $\frac{d}{f} < k_1$.

Case III: $\frac{d}{t} > k_2$, and $\frac{d}{f} > k_1$.

Examining each case separately, we find what modes of failure are possible under it.

Case I: $\frac{d}{t} < k_2$, and $\frac{d}{f} < k_1$. Rivets in row A will shear rather than crush, and those in row B will also shear rather than crush. Hence the only possible modes of failure under this case are G, H, and J. We have thus eliminated three of our six possible modes of failure. A further examination of the remaining three will enable us to eliminate one of them, leaving two.

Case II: $\frac{d}{t} > k_2$, and $\frac{d}{f} < k_1$. Rivets in row A will shear rather than crush, but those in row B will crush rather than shear. Hence the only possible modes of failure under this case are G, H, and L. A further examination of these three will enable us to eliminate one of them, leaving two.

Case III: $\frac{d}{t} > k_2$, and $\frac{d}{f} > k_1$. Rivets in rows A and B will crush rather than shear, and the only modes of failure possible are G, I, and K. A further examination of these three will enable us to eliminate one of them, leaving two.

Let us examine these cases a bit further:

Case I: Possible modes of failure are G, H, and J. Compare G and H

$$G = PtT - dtT$$

$$H = PtT - 2dtT + \frac{\pi d^2 s}{4}.$$

It is evident that H equals G increased by $\frac{\pi d^2 s}{4}$ and decreased by dtT. If the increase equals the decrease then H = G and $\frac{\pi d^2 s}{4}$ = dtT. From this we get

$$\frac{d}{t} = \frac{4T}{\pi s} = k_3 \text{ when } H = G.$$

If $\frac{d}{t} < k_3$, i. e. if $\frac{\pi d^2 s}{4} < dtT$, we know that H will be less than G. Hence the joint can fail only by H or J. (Call this case I(a).) If $\frac{d}{t} > k_3$ we know that G will be less than H. Hence the joint can fail only by G or J. (Call this case I(b).)

By comparing the constants k_1, k_2, and k_3, as given in Table IV, we see that with the usual constants employed k_3 will be less than k_1, but greater than k_2. This being so, it is evident that if $\frac{d}{t} > k_3$ it is also greater than k_2 and would not fall in case I. Hence case I(b) may be disregarded. A summary of this case and its further discussion is as follows:

Case I(a): $\frac{d}{t} < k_2$; $\frac{d}{t} < k_3$; $\frac{d}{f} < k_1$. Failure will be by H or J.

The best pitch to use for this case may be found by equating H and J; hence

$$PtT - 2dtT + as = as + 4aS.$$

$$P = 2d + \frac{4aS}{tT} = P_1.$$

If in this case a pitch less than P_1 be used, the joint will fail by mode H and the efficiency will be $E = \frac{H}{F}$. If, however, a pitch greater than P_1 be used, the joint will fail by mode J and the efficiency will be $E = \frac{J}{F}$.

Case II: Possible modes of failure G, H, and L. Comparing G and H, as under case I, we, of course, arrive at the same conclusion, as follows:

If $\frac{d}{t} < k_3$ then H will be less than G and the joint can fail only by mode H or L. (Call this case II(a).) If $\frac{d}{t} > k_3$ then G will be less than H and the joint can fail only by mode G or L. (Call this case II(b).)

A summary of these two cases and their further discussion is as follows:

Case II(a): $\frac{d}{t} > k_2$; $\frac{d}{t} < k_3$; $\frac{d}{f} < k_1$. Failure will be by H or L.

Examining this for the best pitch to use, we see that it will occur when H = L, or

$$PtT - 2dtT + as = as + 2dtC.$$

$$P = 2d\left(1 + \frac{C}{T}\right) = P_2 \text{ (the best pitch).}$$

If a pitch less than P_2 be used, the joint will fail by mode H and the efficiency will be $E = \frac{H}{F}$. If a pitch greater than P_2 be used. the joint will fail by mode L and the efficiency will be $E = \frac{L}{F}$.

Case II(b): $\frac{d}{t} > k_2$; $\frac{d}{t} > k_3$; $\frac{d}{f} < k_1$. Failure will be by G or L. Examining this for the best pitch to use we see that it will occur when G = L, or

$PtT - dtT = as + 2dtC.$

$$P = \frac{as}{tT} + d\left(1 + \frac{2C}{T}\right) = P_3 \text{ (the best pitch).}$$

If a pitch less than P_3 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F} = 1 - \frac{d}{p}$. If a pitch greater than P_3 be used, the joint will fail by mode L and the efficiency will be $E = \frac{L}{F}$.

Case III: Possible modes of failure are G, I, and K. Comparing G and I we have

$G = PtT - dtT.$

$I = PtT - 2dtT + dfc.$

It is evident that I equals G increased by dfc and decreased by dtT. If the increase equals the decrease then $G = I$ and $dfc = dtT$. From this we get the ratio $\frac{f}{t} = \frac{T}{c} = k_4$.

We have now two subdivisions of this case, as follows:

If $\frac{f}{t} < k_4$ the increase is less than the decrease and $I < G$. (Call this case III(a).)

If $\frac{f}{t} > k_4$ the increase will be greater than the decrease and $G < I$. (Call this case III(b).)

A summary of case III and its further discussion is as follows:

Case III(a): $\frac{d}{t} > k_2$; $\frac{d}{f} > k_1$; $\frac{f}{t} < k_4$. Failure will be by either mode I or mode K.

The best pitch to use for this case may be found by equating I and K; hence

$PtT - 2dtT + dfc = dfc + 2dtC.$

$$P = 2d\left(1 + \frac{C}{T}\right) = P_4 \text{ (the best pitch).}$$

If a pitch less than P_4 be used, the joint will fail by mode I and the efficiency will be $E = \frac{I}{F}$. If a pitch greater than P_4 be used, the joint will fail by mode K and the efficiency will be $E = \frac{K}{F}$.

Case III(b): $\frac{d}{t} > k_2$; $\frac{d}{f} > k_1$; $\frac{f}{t} > k_4$. Failure will be by either mode G or mode K.

The best pitch to use for this case will be found by equating G and K: hence

$$PtT - dtT = dfc + 2dtC.$$

$$P = d\left(1 + \frac{fc}{tT} + \frac{2C}{T}\right) = P_5 \text{ (the best pitch).}$$

If a pitch less than P_5 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F} = \left(1 - \frac{d}{p}\right)$. If a pitch greater than P_5 be used, the joint will fail by mode K and the efficiency will be $E = \frac{K}{F}$.

TRIPLE-RIVETED BUTT-JOINT

Butt-straps of unequal width. (See Fig. 4, Table V, and notations in Table I.)

This joint may fail by any one of the six methods, G to L inclusive, indicated in Table V. The object of this analysis is to determine the conditions under which it will fail by one mode rather than another.

In this joint the outer row, A, contains only rivets that are in single shear, while those of the two inner rows are in double shear.

We have already determined that a rivet in single shear is equally resistant to shear and compression when $\frac{d}{t} = k_1$. We must, however, take into account the butt-strap thicknesses, f and b, as shown in Fig. 4. Usually f is less than t or equal to it. Hence we will use f instead of t because a rivet in single shear will crush in front of the thinner of the two plates. Thus we have $\frac{d}{f} = k_1$ instead of $\frac{d}{t} = k_1$. If $\frac{d}{f} < k_1$, rivets in row A will shear rather than crush. If $\frac{d}{f} > k_1$, rivets in row A will crush rather than shear.

We have also already determined that a rivet in double shear is equally resistant to shear and compression when $\frac{d}{t} = k_2$.

If $\frac{d}{t} < k_2$, rivets in rows B and C will shear rather than crush.

If $\frac{d}{t} > k_2$, rivets in rows B and C will crush rather than shear.

We have the following conditions to consider:

$$\frac{d}{f} < k_1; \ \frac{d}{f} > k_1; \ \frac{d}{t} < k_2; \ \frac{d}{t} > k_2.$$

Apparently two of these conditions may occur at the same time in a joint, but we know that $f < t$ and that $k_1 > k_2$. Hence the number of combinations possible reduces to the three following:

Case I: $\frac{d}{t} < k_2$, and $\frac{d}{f} < k_1$.

Case II: $\frac{d}{t} > k_2$, and $\frac{d}{f} < k_1$.

Case III: $\frac{d}{t} > k_2$, and $\frac{d}{f} > k_1$.

Let us examine each case separately and find what modes of failure are possible under it.

Case I: $\frac{d}{t} < k_2$, and $\frac{d}{f} < k_1$. Rivets in row A will shear rather than crush, and those in rows B and C will also shear rather than crush. Hence the only possible modes of failure under this case are G, H, and J. We have thus eliminated three of our six possible modes of failure. This case will be examined again later.

Case II: $\frac{d}{t} > k_2$, and $\frac{d}{f} < k_1$. Rivets in row A will shear rather than crush, but those in rows B and C will crush rather than shear. Hence the only possible modes of failure for this case are G, H, and L. This case will be examined more in detail later.

Case III: $\frac{d}{t} > k_2$, and $\frac{d}{f} > k_1$. Rivets in all rows A, B, and C will crush rather than shear and the only modes of failure possible are G, I, and K.

A further examination of these three cases will enable us to reduce the modes of failure from three to two, at the same time subdividing the cases.

Case I: Possible modes of failure are G, H, and J. Compare G and H.

$G = (P - d)tT.$

$H = (P - 2d)tT + as.$

It is evident that H equals G increased by $\frac{\pi d^2 s}{4}$, but decreased by dtT. If the increase equals the decrease, H = G, and $\frac{\pi d^2 s}{4} = dtT$. From this we get

$$\frac{d}{t} = \frac{4T}{\pi s} = k_3, \text{ when } H = G.$$

If $\frac{d}{t} < k_3$ we know that H < G. Hence the joint can fail only by H or J. (Call this case I(a).) If $\frac{d}{t} > k_3$ we know that G < H and the joint can fail only by G or J. (Call this case I(b).)

By comparing constants k_1, k_2, and k_3, as given in Table V, we see that with the usual constants employed $k_3 < k_1$, but $k_3 > k_2$. This being so, it is certain that if $\frac{d}{t} > k_3$ it is also true that $\frac{d}{t} > k_2$. This condition could not occur in case I at all. Hence case I(b) does not occur in case I and may be left out of consideration.

A summary of this case and its further discussion is as follows:

Case I: $\frac{d}{t} < k_2$; $\frac{d}{t} < k_3$; $\frac{d}{f} < k_1$. For this case failure may occur only by H or J.

The best pitch for this case may be found by equating H and J.

$PtT - 2dtT + as = as + 8aS.$

$$P = 2d + \frac{8aS}{tT} = P_1.$$

If in this case a pitch less than P_1 be used, the joint will fail by mode H and the efficiency will be $E = \frac{H}{F}$. If, however, the pitch be made greater than P_1, the joint will fail by mode J and

the efficiency will be $E = \frac{J}{F}$.

Case II: Possible modes of failure are G, H, and L. Comparing G and H, as in case I, we reach the same conclusion, as follows, considering L instead of J: If $\frac{d}{t} < k_3$ then H will be less than G and the joint can fail only by mode H or mode L. (Call this case II(a).) If $\frac{d}{t} > k_3$ then G will be less than H and the joint can fail only by mode G or mode L. (Call this case II(b).)

These two subdivisions of case II may now be considered more in detail.

Case II(a): $\frac{d}{t} > k_2$; $\frac{d}{t} < k_3$; $\frac{d}{f} < k_1$. Failure is possible only by H or L. The best pitch occurs when H = L, or

$$PtT - 2dtT + as = as + 4dtC.$$

$$P = 2d + \frac{4dC}{T} = 2d\left(1 + \frac{2C}{T}\right) = P_2.$$

If a pitch less than P_2 be used under this case, the joint will fail by mode H and the efficiency will be $E = \frac{H}{F}$. If a pitch greater than P_2 be used, the joint will fail by mode L and the efficiency will be $E = \frac{L}{F}$.

Case II(b): $\frac{d}{t} > k_2$; $\frac{d}{t} > k_3$; $\frac{d}{f} < k_1$. Failure will be by G or L. The best pitch is found by equating G and L.

$$PtT - dtT = as + 4dtC.$$

$$P = d\left(1 + \frac{4C}{T}\right) + \frac{as}{tT} = P_3.$$

If a pitch less than P_3 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F} = 1 - \frac{d}{p}$. If a pitch greater than P_3 be used, the joint will fail by mode L and the efficiency will be $E = \frac{L}{F}$.

Case III: Possible modes of failure are G, I, and K. Comparing G and I we have

$$G = (P - d)tT.$$

$I = (P - 2d)tT + dfc.$

If $G = I$ we have $dTt = dfc$ and $\frac{f}{t} = \frac{T}{c} = k_4$. If $\frac{f}{t} < k_4$ then $I < G$ and the joint may fail by I or K. (Call this case III(a).) If $\frac{f}{t} > k_4$ then $G < I$ and the joint may fail by G or K. (Call this case III(b).)

Considering these two subdivisions in detail we have

Case III(a): $\frac{d}{t} > k_2$; $\frac{d}{f} > k_1$; $\frac{f}{t} < k_4$. Failure is by I or K.

If we equate I and K we find the best pitch as follows:

$PtT - 2dtT + dfc = dfc + 4dtC.$

$$P = 2d\left(1 + \frac{2C}{T}\right) = P_4.$$

If a pitch less than P_4 be used, the joint will fail by mode I and the efficiency will be $E = \frac{I}{F}$. If a pitch greater than P_4 be used, the joint will fail by mode K and the efficiency will be $E = \frac{K}{F}$.

Case III(b): $\frac{d}{t} > k_2$; $\frac{d}{f} > k_1$; $\frac{f}{t} > k_4$. Failure will be by G or K.

The best pitch for this case occurs when $G = K$, or

$PtT - dtT = dfc + 4dtC.$

$$P = d\left(1 + \frac{4C}{T} + \frac{fc}{tT}\right) = P_5.$$

If a pitch less than P_5 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F} = 1 - \frac{d}{p}$. If a pitch greater than P_5 be used, the joint will fail by mode K and the efficiency will be $E = \frac{K}{F}$.

QUADRUPLE-RIVETED BUTT-JOINT

Butt straps of unequal widths. (See Fig. 5, Table VI, and notations in Table I.)

This joint may fail by any one of the eight modes, G to N inclusive, indicated in Table VI.

The two outer rows of rivets, A and B, are in single shear, while those of the two inner rows, C and D, are in double shear. By reasoning developed in previous paragraphs*, we may state the following:

If $\frac{d}{f} < k_1$ rivets in rows A and B will shear rather than crush. If $\frac{d}{f} > k_1$ rivets in rows A and B will crush rather than shear. And:

If $\frac{d}{t} < k_2$ rivets in rows C and D will shear rather than crush. If $\frac{d}{t} > k_2$ rivets in rows C and D will crush rather than shear.

By reasoning similar to that at the top of page 307 we may at once write the three main cases.

Case I: $\frac{d}{t} < k_2$, and $\frac{d}{f} < k_1$.

Case II: $\frac{d}{t} > k_2$, and $\frac{d}{f} < k_1$.

Case III: $\frac{d}{t} > k_2$, and $\frac{d}{f} > k_1$.

Let us examine each case separately and find what modes of failure are possible under it.

Case I: $\frac{d}{t} < k_2$; $\frac{d}{f} < k_1$. Rivets in rows A and B will shear rather than crush. Rivets in rows C and D will also shear rather than crush. Hence of the eight modes of failure tabulated only G, H, J, and L are possible under this case.

Case II: $\frac{d}{t} > k_2$; $\frac{d}{f} < k_1$. Rivets in rows A and B will shear rather than crush. Rivets in rows C and D will crush rather than shear. Hence under this case the four possible modes of failure are G, H, J, and N.

Case III: $\frac{d}{t} > k_2$; $\frac{d}{f} > k_1$. Rivets in all rows will crush rather than shear. Hence the possible modes of failure for this case are G, I, K, and M.

A further examination of these cases will enable us to sub-

*Top of p. 280.

divide them so that but two modes of failure are possible under each division.

Case I: Possible modes of failure are G, H, J, and L. Compare G, H, and J.

$G = (P - d)tT.$
$H = (P - 2d)tT + as.$
$J = (P - 4d)tT + 3as.$

It is evident that H equals G increased by $as = \frac{\pi d^2 s}{4}$, but decreased by dtT. If the increase equals the decrease we get $\frac{d}{t} = \frac{4T}{\pi s} = k_3$ when $H = G$.

Comparing G and J we arrive at exactly the same result, i. e. when $\frac{d}{t} = k_3$ we have $H = G = J$.

If $\frac{d}{t} < k_3$ the increase will be less than the decrease and J will be less than either G or H. For this condition the only possibles modes of failure for the joint are J and L. (Call this case I(a).)

If $\frac{d}{t} > k_3$ the decrease will be less than the increase and G will be less than either H or J. For this condition the only possible modes of failure for the joint are G and L. (Call this case I(b).)

For the reason given near the middle of page 308 we see that with the usual unit stress values we need not further consider case I(b).

A summary of case I and its further discussion is as follows:

Case I: $\frac{d}{t} < k_2$; $\frac{d}{t} < k_3$; $\frac{d}{f} < k_1$. The possible modes of failure under this case are J and L.

The best pitch for this case may be found by equating J and L.

$PtT - 4dtT + 3as = 3as + 16aS.$

$P = 4d + \frac{16aS}{tT} = P_1$ (the best pitch).

If in this case a pitch less than P_1 be used, the joint will fail by mode J and the efficiency will be $E = \frac{J}{F}$. If, however,

a pitch greater than P_1 be used, the joint will fail by mode L and the efficiency will be $E = \frac{L}{F}$.

Case II: Possible modes of failure are G, H, J, and N. By comparing G, H, and J, as at the top of page 312, we arrive at the same conclusion and hence subdivide this case as follows:

If $\frac{d}{t} < k_3$ failure under this case will be by J or N. (Call this case II(a).) If $\frac{d}{t} > k_3$ failure under this case will be by G or N. (Call this case II(b).)

Considering these subdivisions more in detail we have

Case II(a): $\frac{d}{t} > k_2$; $\frac{d}{t} < k_3$; $\frac{d}{f} < k_1$. Possible modes of failure are J and N.

To find the best pitch for this case we equate J and N as follows:

$$PtT - 4dtT + 3as = 3as + 8dtC.$$

$$P = 4d\left(1 + \frac{2C}{T}\right) = P_2 \text{ (the best pitch)}.$$

If a pitch less than P_2 be used, the joint will fail by mode J and the efficiency will be $E = \frac{J}{F}$. If, however, a pitch greater than P_2 be used, the joint will fail by mode N and the efficiency will be $E = \frac{N}{F}$.

Case II(b): $\frac{d}{t} > k_2$; $\frac{d}{t} > k_3$; $\frac{d}{f} < k_1$. Possible modes of failure are G and N.

To find the best pitch for this case equate G and N as follows:

$$PtT - dtT = 3as + 8dtC.$$

$$P = d\left(1 + \frac{8C}{T}\right) + \frac{3as}{tT} = P_3 \text{ (the best pitch)}.$$

If a pitch less than P_3 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F} = 1 - \frac{d}{p}$. If, however, a pitch greater than P_3 be used, the joint will fail by mode N and the efficiency will be $E = \frac{N}{F}$.

Case III: Possible modes of failure are G, I, K, and M

Compare G, I, and K.

$G = (P - d)tT.$

$I = (P - 2d)tT + dfc.$

$K = (P - 4d)tT + 3dfc.$

If $G = I = K$ we will have $dtT = dfc$, or $\frac{f}{t} = \frac{T}{c} = k_4$.

If $\frac{f}{t} < k_4$ then $dfc < dtT$ and K will be less than either G or I. Hence under this condition the joint can fail only by K or M. (Call this case III(a).)

If $\frac{f}{t} > k_4$ then G will be less than either I or K. Hence under this condition the joint can fail only by G or M. (Call this case III(b).)

A summary and further consideration of these subdivisions follows:

Case III(a): $\frac{d}{t} > k_2; \frac{d}{f} > k_1; \frac{f}{t} < k_4$. Possible modes of failure are K and M.

To find the best pitch equate K and M as follows:

$PtT - 4dtT + 3dfc = 3dfc + 8dtC.$

$P = 4d\left(1 + \frac{2C}{T}\right) = P_4$ (the best pitch).

If a pitch less than P_4 be used, the joint will fail by mode K and the efficiency will be $E = \frac{K}{F}$. If, however, a greater pitch than P_4 be used, the joint will fail by mode M and the efficiency will be $E = \frac{M}{F}$.

Case III(b): $\frac{d}{t} > k_2; \frac{d}{f} > k_1; \frac{f}{t} > k_4$. Possible modes of failure are G and M.

To find the best pitch equate G and M as follows:

$PtT - dtT = 3dfc + 8dtC.$

$P = d\left(1 + \frac{3fc}{tT} + \frac{8C}{T}\right) = P_5$ (the best pitch).

If a pitch less than P_5 be used, the joint will fail by mode G and the efficiency will be $E = \frac{G}{F} = 1 - \frac{d}{p}$. If a greater pitch than P_5 be used, the joint will fail by mode M and the efficiency will be $E = \frac{M}{F}$.

DISCUSSION

MR. KENNETH H. TALBOT:* I should like to ask if the methods presented would give 6½ inches as the most economical spacing. You stated that this spacing was assumed in the example you worked. (See page 289.)

MR. HOWARTH: This can be determined by referring to Table X. In case II(b) we find that $P_3 = \frac{0.8a}{t} + 7.91d = 7.5$ inches. The best pitch in that case would be 7.5 inches instead of 6.5, so evidently the pitch assumed was less than the one that would give the greatest efficiency. This table also shows that when the assumed pitch is less than the best pitch the efficiency for this case is shown by $E = \frac{G}{F}$. This checks up with the calculations made for the example.

MR. EDWARD GODFREY:† Do you find any justification for taking the full driven diameter of the rivet? In structural work we always take the nominal diameter of the rivet. Second, is there any justification for using 95 000 pounds as the compressive strength of the rivet, when the material used is only 55 000-pound steel?

In the outside row of rivets I believe the pitch is 6½ inches. Wouldn't that be hard to calk on a ⅜ plate?

MR. HOWARTH: Answering your last question first; the 6½-inch pitch is for the outside row of rivets. Calking would be done against the edges of the outer butt-strap where the pitch is 3¼ inches.

With regard to the values of the constants; I have seen a good many recommended and in each case the set recommended for riveted-joint calculations appears to be adjusted so as to give results which agree with tests which have been made on the joints. In most instances that have come to my attention the unit crushing resistance is taken at 95 000 or thereabouts for boiler joints. In some cases I have seen it higher than that, but very seldom lower.

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†Structural Engineer, Pittsburgh

In answer to your first question; I would say that, if you drill a hole for a rivet, the effective width of plate you have left must necessarily be the distance between the two sides of the *holes*—not the sides of the rivets. You should figure the tearing strength of the plate from the actual section of the plate. When the rivet is driven it may not always swell up and fill the hole, but it is supposed to. It will with short rivets. So you are really justified in figuring the driven diameter.

Mr. Edward Godfrey: It does not show up well in comparison with our standards for structural design.

Mr. Howarth: Perhaps I ought to apologize to somebody for putting this paper before a structural section, because I made the study particularly in connection with boiler joints. In structural work you would be interested primarily in only two joints that I have discussed—the lap-joint, and the butt-joint in which the pitch is the same in all rows and the straps are of equal width.

Mr. Edward Godfrey: We like to make comparisons between standards of design of boilers and structural work. We do not often come across boiler standards, but things like this puzzle us—why it is that boiler design, where so much depends on the strength of the joint, is not nearly so safe, comparatively, as structural design.

Mr. Howarth: In what way?

Mr. Edward Godfrey: We add ⅛ inch to the diameter of the rivet for the hole in finding the net section.

Mr. Howarth: In figuring the tearing strength of the plate you assume that the metal around the hole is no good for a certain distance out?

Mr. Edward Godfrey: Yes, and we use only the nominal diameter of the rivet, even in drilled holes, when figuring the rivet strength.

MR. H. D. JAMES:* Cannot this be explained by a variation in the factor of safety employed?

MR. HOWARTH: I believe the Code requires a factor of safety of five on the ultimate values.

MR. EDWARD GODFREY: We do not use the ultimate; we use units for safe strength of the rivet in bearing and shear.

MR. H. D. JAMES: Those units are based on the elastic limit. There are two ways of figuring—one on the ultimate and the other on the elastic limit, taking the factor of safety on either of these values.

MR. EDWARD GODFREY: Usually where we have 16 000 pounds tension on the steel we have 10 000 shear on the rivet and 20 000 bearing. This is 25 per cent. more in bearing than in tension. In comparison, if we use 55 000 for the ultimate, we would use 25 per cent. more than that—which would be less that 70 000—for the bearing on the rivet, which you call compression; and our shear would be half of that.

MR. H. D. JAMES: Are these figures based on the results of tests?

MR. EDWARD GODFREY: To a certain extent, they are.

MR. H. D. JAMES: I thought perhaps they represented arbitrary decisions rather than test data.

MR. EDWARD GODFREY: Tests demonstrate the strength of our joints, but there is also some rational basis.

MR. HOWARTH: I understand it is quite rational, especially in connection with punched holes, to assume that the plate is practically worthless as far as strength is concerned for about 1/8 inch around the hole.

MR. EDWARD GODFREY: That is the reason we add practically 1/16 inch to the diameter of the punched hole—1/8 more than the diameter of the rivet, for there is 1/16 inch clearance.

*Assistant to Manager of Engineering, Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa.

Mr. Howarth: For uniformity you might do that whether the hole is punched or drilled.

Mr. Edward Godfrey: Punched or drilled or reamed. It is all the same in structural work. We take the nominal diameter of the rivet, always; and in the net section we take out 1/8 inch more than the nominal diameter.

Mr. Howarth: As far as the method given in this paper is concerned I do not see why the principles could not be applied equally well to your work.

Mr. Edward Godfrey: You would have to make allowance for that 1/8 inch in diameter.

Mr. Howarth: I can not say, offhand, just how that would work out but it does not look prohibitive.

Mr. Edward Godfrey: You would have to have some revision to apply to tank work. Tank plates are only punched, not reamed.

Mr. H. D. James: Am I correct in understanding that this is the new A. S. M. E. Code?

Mr. Howarth: The new Code standard method of figuring boiler-joints has been taken as a basis for this paper.

Mr. Edward Godfrey: Was any marked change in standards made in adopting this new Code?

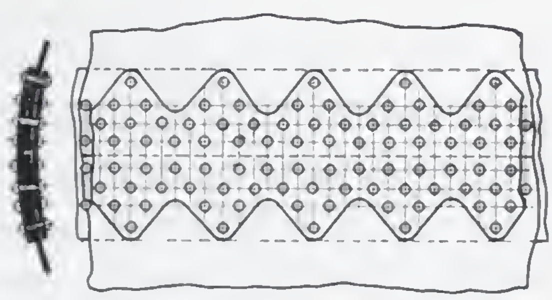
Mr. Howarth: Not that I know of in riveted joints.

Mr. Edward Godfrey: What about single lap-joints? Are they permitted?

Mr. Howarth: I believe they are in some cases in ring seams, and for some low-pressure work.

Mr. Edward Godfrey: Are there not a good many failures of boilers in single lap-joints?

Mr. Howarth: I believe a good many of the older boilers were made with lap-joints, and the older boilers, of course, are the ones that are failing, due to corrosion. There are certain actions in the lap-joint which are criticised by boiler designers. In fact, all the joints I have shown are open to criticism because they are unsymmetrical, with the exception of the one with butt-straps of equal width. In order to get a high efficiency with that type of joint, and be able to calk it, some engineers recommend a saw-tooth joint. They notch the outer butt-strap so the edges will be properly supported by the rivets, to make calking effective. (See Fig. 11.)



BUTT AND DOUBLE STRAP-JOINT
OF THE SAW-TOOTH TYPE

Fig. 11.

Mr. Edward Godfrey: There is a tendency to bend the rivet in a single lap-joint.

Mr. Howarth: Yes, and the same thing is true with the butt-joint with straps of unequal width.

The method adopted by the Code Committee was undoubtedly that which seemed best to the Committee. Some method had to be adopted for joint calculations. There was probably no simple method available that was not open to objection. In your own work there are no doubt many stress calculations which you make in only an approximately rigorous way. You take care of errors in your analysis by using low stresses.

Mr. Edward Godfrey: Well, the stresses in the boilers are very definite.

Mr. Howarth: They are fairly definite. In this paper I have been assuming that a plate pulls equally on all similar rows of rivets. I have neglected the stretching of the plate. This is not absolutely rigorous. Actually the metal at the inner rows gets less stress and that at the outer row more stress than the method given in this paper would show.

Mr. Edward Godfrey: I never considered that those things were worth bothering about. But there are big things—the effective diameter of the rivet; the real net section of the plate; the unit of bearing; and bending of the rivet—there are four features of design of boiler joints that, to my mind, are big; and I do not think they are very well taken care of in that standard.

Mr. Howarth: As I understand it the Boiler Code Committee is taking care of the faults of simple joints by permitting them for low-pressure work; for high pressures it requires the more complex joints, in which the plate is not subject to so much bending action. One engineer who has written a great deal on the subject of riveted joints—Mr. Dean of Boston—is in favor of the saw-tooth joint and equal width butt-straps, because that type of joint does not bend the main plates.

Mr. Edward Godfrey: I wonder if any boiler was ever made that way.

Mr. Howarth: I think that method is used by Mr. Dean.

Mr. Edward Godfrey: And are they calked inside and outside?

Mr. Howarth: No. The outer strap is made saw tooth so the rivets will properly support the edge when it is calked. The inner strap is not calked. These joints have not been officially adopted by the Boiler Code Committee. They are mentioned and shown, but nothing is said about the way of figuring their efficiency. It would be by standard methods, with rivets figured in double shear. There has been a lot of argument over this type of joint and it may be more widely used in the future.

MR. ALBERT KINGSBURY:* The thing that impresses me in the paper is the manner in which the curves enable us to visualize the whole problem. To any one who remembers the difficulties he has had in getting something tangible on this subject, these curves are very instructive. I think Mr. Howarth has shown for the first time a set of curves which will enable one to get a grasp of what the formulas mean.

It does not seem to me that it is the intention in the paper to fix allowable values of moduli of rupture, such as 44 000 pounds in shear and 95 000 in compression. The allowable stresses vary with the material and each manufacturer must determine the proper values for his special material.

With regard to the details of the stresses in the joints, we should remember that these joints are, on the whole, designed in a more or less empirical fashion; that in none of the formulas is there any attempt to express the real maximum stresses in the parts; and that average stresses only are dealt with. Now the actual maximum stresses may be very much greater than the average. For example: It has been shown both by analysis and by experimental proof that if, in a plate of considerable width subject to a uniform pull across its whole width, a small hole is made through the plate, the stress at the hole goes up to three times the average stress in the plate. Thus, the plate between rivet holes, instead of being subject to uniform stress, may have a maximum stress which may be three times the average; and very probably the action of the rivet in pressing against the side of the hole still further increases the local tensile stress at the hole. This consideration emphasizes the importance of drilling the holes rather than punching them; or of reaming the holes, if punched, in order to remove the metal that may have been cracked by punching.

I have no doubt there are other important but more or less obscure stresses in the plates and rivets, of which we have no very exact knowledge, which it would be necessary for us to know in order to have a complete and final analysis of the joints.

*Consulting Engineer, Pittsburgh.

Mr. Paul Whitman:* In answer to the question of Mr. James with reference to the relative factors of safety for steel construction, commonly used for structural work, as compared with the factors for plate and boiler work, I would state that for the general run of steel construction for buildings it is common practice to use a factor of safety of four. In designing steel members for crane runways, or other parts of structures subjected to moving live loads, we either use a straight factor of safety of five, or use a factor of four after we have increased our stresses by adding 25 per cent. additional for impact; which is equivalent to using a factor of five. In railway bridge work, for extra long compression members, more than 25 per cent. additional stress for impact is often used. The ultimate tensile strength of steel plates varies from 65 000 to 55 000 pounds per square inch. In case the tension forces to which the material is subjected act parallel to the direction in which the plate passes through the rolls in the plate mill, we can use the former figure of 65 000 pounds; but if the external forces act perpendicular to this direction we must use the latter figure of 55 000 pounds per square inch. As the external surface of the plate gives no indication of which way it passed through the rolls, we must guard against its being placed the weaker way in the completed structure by using the lower ultimate value of 55 000. The factor of safety of four on 55 000 pounds ultimate, commonly used in plate work, is, therefore, approximately equivalent to a factor of five on the larger ultimate value. Working conditions in the use of structural steel shapes, on the contrary, insure this being used in such a way that we can safely use the larger ultimate value.

In boiler work I understand the common practice is to use a factor of safety of five, in comparison with the factor of four on 55 000 pounds ultimate used in plate work. This means a factor of six compared with the factor of four used for ordinary structural building work. In boiler work the rivets are usually driven with hydraulic riveters and other precautions are taken to insure a better grade of workmanship in the shop than is ordinarily obtained for the usual grade of structural and plate work.

*Engineer, Riter-Conley Co., Pittsburgh.

In summing up I think, therefore, we can safely say that the relative factors of safety in common use for steel construction are relatively from a general standpoint: four for structural work; five for plate work; and six for boiler work. Before closing my remarks I would like to compliment the author on the excellency of his paper. It will be a valuable contribution to our PROCEEDINGS. There is not much literature on the subject. Most shops have evolved certain standard pitches—based on the diameter of the rivets used—which they have found by experience give them the least trouble in punching and working the various thicknesses of steel plates which they fabricate. They have not taken time to investigate, or calculate the proper pitch of rivets to give the maximum efficiency of the material involved.

MR. F. L. EGAN:* Mr. Howarth's paper, it appears, applies to a class of riveted work distinctly different from standard structural riveting; that is, boilers, pressure tanks, and mechanical details subject to alternate strains of opposite signs and at comparatively high frequency—from 18 to 300 cycles per minute.

The structural designer sometimes attempts to use his structural "standards" as a universal rule by which all and any riveted construction is to be measured, and when so applied to the class of work mentioned above it frequently produces expensive failures. Structural standards usually make no allowance for impact and the designer uses his discretion in selecting and adding an arbitrary or empirical percentage to the loads. Structural hot riveting in rough-punched holes, one-sixteenth inch larger than nominal size of rivets, will not successfully stand alternate strains of opposite signs at comparatively high frequency, unless extremely low unit stresses are used both in shear and bearing values, in which case the resultant high factor is more a "factor of ignorance" than a factor of safety.

Due to one of, or to combinations of, the following reasons—1. holes are tapered; 2. holes do not align even after drifting; 3. rivet is not driven to fill; 4. rivet shrinks so as to reduce section at center—when the joint is subjected to a strain of static load plus impact, which exceeds the frictional resistance of the joint

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plus actual surface in shear and bearing, a displacement is started which is culminative when joint is subjected to alternate strains of opposite signs.

The mechanical details I have in mind are similar to the forged and bolted locomotive frames in use until recent years. The bolts were turned taper, 3/32 inch per foot, fitted to reamed holes and driven as tight as the bolt would stand. One and one-quarter inch diameter was the most common size and we had a shop-made cannon to shoot them out. I have seen these bolts offset as much as ⅛ inch after several years of service,—making it necessary to drill them out—when loaded to a unit shearing stress of 3700 pounds per square inch due to total piston thrust of engine.

A higher carbon steel was used for the bolts, which caused trouble from breakage of bolts. No trouble was experienced with crushing of material.

A similar case is a plate girder used in the same sort of service as the locomotive frame—that is, as a frame for a twin tandem-compound engine with cranks set 90 degrees apart—They usually ran about 30 inches wide in the web at the center tapering to about 12 inches wide at the ends. Web of ⅜-inch plate with two 6- by 6- by ½-inch angles and a 10-inch wide by 13/16-inch thick plate, top and bottom. These beams were formerly made of the finest of white or long-leaf pine, almost impossible to secure now at any price. The first steel girders failed —one case in less than thirty days service—and it is only by reducing unit fiber stresses to a very low value and insisting on the best of shop work that success is attained. Holes drilled from the solid, and hydraulically cold-pressed rivets should make a good job.

Regarding boiler joints, good modern practice is to have all rivet holes drilled from the solid, with parts bolted into position and then taken apart and all burrs and rough sharp edges removed before riveting. For several reasons it is not strictly correct to use a factor of safety in regard to boiler work. An instance is the joint Mr. Howarth has calculated on the blackboard. For best efficiency he gets 7½-inch pitch, or a half pitch, or a calking distance on outside butt-strap of 3¾ inches. You will

find that the United States marine law allows you a calking distance of about 3¼ inches on this thickness of plate, which cuts your pitch to 6½ inches.

It is very interesting to attempt to construct a boiler to fulfill all the requirements of the A. S. M. E. Code as applied to boilers; the United States marine law; and the Hartford Inspection and Insurance Company. You will find factors of safety which vary from six to barely three, and requirements which are directly contradictory.

Speaking of the saw-tooth joint, I worked on its development just eighteen years ago. It was a hobby of Mr. Mathew Moore, general superintendent of the Atlas Engine Works, Indianapolis, Ind. The Atlas Works at that time were averaging eighteen boilers to a 10-hour day. A machine was designed and built to mill the serrated edges on three butt shapes at once. We soon found that it was not a commercial proposition as we could either thicken up the butt-straps a slight amount or rearrange our joint and use a straight-edge butt-strap and get the same result at less expense and at less hazard in shop work. Conditions to-day may make it feasible.

Mr. Paul Whitman: I would like to ask Mr. Howarth what thickness he would recommend for the butt-plates in boiler work. Would he make both butt-straps the same thickness, or would he advocate making them different thicknesses? In designing the butt-straps for large water tanks it is the practice of my firm to make the combined thickness of the butt-straps not less than 1¼ times the thickness of the shell plate. Concerning the relative thicknesses of the two butt-plates, we make the inside butt-plate 1.4 times the thickness of the outside butt-plate; provided that no butt-plate shall be less than half the thickness of the shell plate.

Mr. Howarth: I understand that in boiler work they are frequently made of the same thickness as the main plate and in some cases less. I could not find by reference to the Code the exact values recommended but, in considering the modes of failure possible, the Committee says that joints designed according

to its rules will not fail by tearing the two butt-straps. That means that the butt-straps must be fairly thick—possibly three-quarters of the thickness of the main plate, or more.

The thickness of butt-strap should depend on the calking pitch, as mentioned by a previous speaker. If you think the strap is not thick enough to calk at a certain pitch, you must increase the thickness or reduce the pitch. When I stated that 7½ is a more efficient pitch than the 6½ inches given in the example, I did not mean to recommend that particular pitch, but to call attention to the fact that 7½ inches will give a better efficiency that 6½. If we had a set of curves that showed the properties of that joint we might even find that we could use a better rivet than 13/16. (Since the presentation of the original paper, the curves for this joint have been drawn. See Fig. 9.)

MR. PAUL WHITMAN: In answer to a rather disparaging remark about a plate girder not being able to stand up to the calculations, I think the trouble must have been caused by poor shop work, rather than by any fault or error of the calculations. It is my opinion that the failure was caused by the rivets not properly filling the holes. Successive shocks and other hammering cause such rivets to work loose, and to fail ultimately, under loads much less than they have previously withstood. In plate girders, built up of thick flange angles and cover plates, it is very seldom that the rivets connecting the angles and plates completely fill the holes. Any one who has ever had occasion to knock out such rivets can very readily testify to this.

MR. GEORGE H. DANFORTH, CHAIRMAN: I think we all realize that in a lot of structural work and bridge work we do not get the rivets to fill the holes as fully as is done in ordinary boiler work. The boiler maker realizes that he must have his work tight, because anybody can see a jet of water coming out around a boiler rivet but you do not see it in structural work.

MR. H. D. JAMES: Mr. Kingsbury has pointed out the important point in the paper; namely, the methods used to visualize the best solution of a riveting problem.

The discussion has been based largely upon the different practices which exist in various branches of the profession and has to do largely with factors of safety and workmanship.

Material should not be strained beyond its elastic limit. The amount of stress that any form of steel will stand depends to a considerable extent upon the heat treatment which it received during the fabricating period. The relation between the elastic limit and the ultimate fiber strength is also affected by the heat treatment. It is probable that a good many differences in tests could be explained if we had a history of the samples tested and knew at just what point the elastic limit was reached.

In a discussion of the proper design of riveted joints, the engineer is endeavoring to form a balance between the strength of the joint and the cost of its manufacture. This ratio changes from time to time and is controlled to a large extent by the particular shop facilities at his command. During the past year manufacturing conditions have been changing so rapidly that it has become very difficult to maintain the proper balance between design and cost. The paper of the evening has demonstrated a method of preparing charts which will materially simplify the problem and enable us to make our designs follow shop conditions more closely.

MR. WILLIAM KENT:* The author has done a valuable service for designers of riveted joints in showing how the usual calculations may be simplified and shortened. I wish he had gone further and given us a formula for the best diameter of rivet, and a handy table showing for given thicknesses of plate and strength of material, the diameter of rivet, the pitch and the efficiency.

I have made a calculation, based on the data given in Table X, of a triple-riveted butt-joint with unequal butt-straps for ½-inch plate. The rivet hole diameters were selected so as to come within the limits of cases I, II(a), II(b), and calculations have been made for four possible methods of failure, C, H, J, and L. The results are shown below ($t = 0.5$; $f = 0.4$; $\frac{d}{f} < 2.75$ in all cases. $T = 55\,000$; $S = 44\,000$; $C = 95\,000$).

*Consulting Engineer, Montclair, N. J.

Case I	Case II(a)	Case II(b)
$\frac{d}{t} < 1.375$	$\frac{d}{t} > 1.375$, < 1.592	$\frac{d}{t} > 1.592$
$d < 0.687$	$d > 0.687, < 0.796$	$d > 0.796$

	Case I	Case II(a)	Case II(b)	
d taken at..............	0.65	0.75	0.85	1.00
a, sq. in.	0.3318	0.4418	0.5674	0.7854
Best pitch in............	5.547	6.683	7.631	9.167
$G = (P - d)tT$.......	134 667	163 157	186 477	223 842
$H = (P - 2d)tT + as$.	131 691	161 971	188 068	232 660
$J = 9aS$	131 393	174 953	224 690	320 112
$L = aS + 4d + C$.....	138 099	161 939	186 466	225 568
Unperforated plate, $F = PtT$	152 542	183 782	209 852	252 092
Efficiency, per cent.	86.1	88.1	88.96	88.8

The result is rather surprising. With rivet hole diameters varying all the way from 0.65 to 1.00 inch and corresponding pitches from 5.55 to 9.17 inches the efficiencies vary only between 86 and 89 per cent., and for diameters from 0.75 to 1 inch and pitches 6.68 to 9.17 inches the whole range of efficiency is within one per cent.

Is there any reason why we should not adopt ⅞ inch as the standard diameter of rivet hole for ½-inch boiler steel plate with $T = 55\,000$, $S = 44\,000$, $C = 95\,000$? The corresponding pitch, according to the formula $P_3 = \frac{0.8a}{t} + 7.91d$, is 7.883, practically 7⅞ inch, which may be taken as the standard pitch. The strength of this joint by method G is 192 500 pounds; by method L, 192 700 pounds. The efficiency is 88.9 per cent.

Before a standard is finally adopted, however, we should have some more experimental knowledge concerning the shearing and crushing strength of the material. The tensile strength of good fire-box and boiler-plate steel may safely be taken at 55 000 pounds, but do we know that the shearing strength of the rivets is 44 000 and that it is safe to take the same figure

whether the rivet is in single or in double shear? Do we know that the crushing strength of steel of 55 000 pounds tensile strength is 95 000 pounds, and do we know that this figure is independent of the thickness? What is crushing strength? Is it the load, just beyond the elastic limit in crushing, at which the metal will begin to flow, or is it the load that will produce a definite amount of shortening of the test piece?*

The pitch of 7⅞ inches may be objected to as being difficult to calk so as to make a steam-tight joint. This should be investigated before standard dimensions are adopted.

The questions raised above might be referred to the United States Bureau of Standards for investigation after the end of the war.

AUTHOR'S CLOSURE: No doubt handy tables can readily be made up as suggested by Mr. William Kent, from the results of this analysis. It seems to the author, however, that the most satisfactory form in which the results could be arranged would be a set of curves similar to those given in Fig. 6–10. If a set were to include every desirable thickness of plate it would be somewhat voluminous, though perhaps not prohibitively so. The simplest set of curves would not show pitches and rivets directly, but only as ratios to plate thicknesses, as suggested elsewhere in this paper (page 292). This method would not work out quite so simply for the joints having butt-straps of unequal widths. A set would be required for each ratio of inner strap thickness to main plate thickness.

Mr. Kent is rather surprised to find the efficiencies of the triple-riveted butt-joint of Fig. 5 so uniform for such widely different rivet diameters. The curves in Fig. 9 show graphically the properties of the joint, though the plate thickness is different. The efficiencies he calculated are at the apexes of the curves. For greater pitches the efficiencies fall off rapidly, but for lesser pitches they fall off slowly. This fact argues that for this type of joint the pitch used should not exceed the best pitch for the rivet. Preferably it should average somewhat less.

*For a discussion of compressive strength see Kent's "Mechanical Engineers' Pocket-Book." Ed. 9, p. 281.

The author understands that the calking pitch should bear a reasonable relation to the thickness of plate that is calked. That being the case it is advisable to fix that rule first and determine the best rivet afterwards.

The author agrees with Mr. Kent's suggestion that the matter of the relative values of tension, shear, and compression for riveted joints may be profitably investigated further. It is evident from the remarks by Mr. Godfrey that structural and boiler practice do not agree in this respect.

It is not the intention of this paper to recommend or defend any set of ultimate stress values, nor even to defend the system of calculating riveted joint efficiencies adopted by the Boiler Code Committee of the American Society of Mechanical Engineers. The author has endeavored simply to interpret that system first by analysis and next by means of curves that show the joint properties at a glance. It is hoped that the subject may by this means be made clearer to the minds of those interested in the design of riveted joints.

The author wishes to thank the gentlemen who have been kind enough to participate in the discussion of the paper. It is regretted that the few advance copies which were sent out did not contain all of the figures and explanations that are contained in the complete paper.

PRODUCER GAS, ITS MANUFACTURE AND USE

By Dr. C. S. Palmer*

The subject of producer gas should perhaps have a little preliminary definition. If you look in most of the books you will find that the term "producer gas" is intended, by a good many of the old books, to cover the production of CO by the partial oxidation of carbon; and the term "water-gas" is intended to describe the gas resulting from the action of water on hot coal, which consists largely of CO plus H_2. In the later practice, however, as you all know, and in the ordinary producer, both air and steam are passed over hot coal; and therefore there must be, as a result, more or less of a mixture of these two. And we find that the ordinary producer gas bears this out as though the two reactions did occur. There is usually twice as much CO_2 as there is H_2, indicating both the old producer-gas reaction—the incomplete oxidation of the C—and the water-gas reaction. The Germans call this gas "mischgas" or mixed gas; and some have tried to introduce this term into American books and practice. When you come to think that most of the producer gas is used in great steel- and glass-works, where air and steam are both thown in, why shouldn't we drop the old distinction and call producer gas that gas which is produced by the ordinary producer in the largest American practice? Our ordinary producer gas, then, consists roughly of 10 or 12 per cent. of H_2, 20 to 25 per cent. of CO, and the remainder very largely N_2 and some CO_2. So much for the definition of the term as we shall use it this evening.

I should perhaps say a word about my own experience. Ten years ago in company with some friends I went into the gas-producer and gas-engine business in Boston. We were very young indeed and there were very few books to help us. In a word it meant this: In the small factories, where ordinary steam

*Mellon Institute, University of Pittsburgh, Pittsburgh.

power is costing four to six pounds of coal per horsepower-hour, roughly, the producer and the gas-engine could give them a horsepower-hour for about one pound of coal, this coal being anthracite. We did not realize at that time that the story of producer gas with you people, using soft coal, was a very old story. We did not know enough to come down and learn from your experience. And we had great difficulties even with anthracite. And let me tell you that is a mere picnic as compared with the use of bituminous coal. I visited almost every producer-gas plant in New England, and found only one plant using bituminous coal. I think they were using Westinghouse engines at the American Steel & Wire plant at Worcester, Mass. So we went back to our anthracite, and one of the engineers in one of the large packing-houses in Somerville, Mass., told me this: "You have a very good thing, but you have no engineers to stand back of you if anything goes wrong with the gas-engine." That was true. The engineers and mechanics were past-masters in steam, but if anything happened to the producer or engine they "had fits."

The trouble with the anthracite producer, as I look at it, is that in most small plants it does not run on a 24-hour circuit. It is supposed to start at 7 a.m. That means that it must be warmed up about 4 a.m., and it probably gets into good running shape some time in the afternoon, and is shut down again at night. I judge we would have had comparatively little trouble if we could have run on a continuous system. We heard of producer gas being used in steel plants and we got as far as Buffalo, and saw the remarkable engines using furnace gas—waste blast-furnace gas. We also heard of the new works at Gary, Ind.

And here I want to call your attention to some of the remarkable things about producer gas. I do not know any great subject which has been so much abused in the house of its friends as producer gas has in the books of the authorities. In general the gas-producer books do not seem to have been written by men who knew the subject. While I know very little about the subject, yet I have seen enough of practical engineering to believe that as a rule, unless it can not be done in some other way, nobody should be allowed to write a book on an engineering subject, without having had several years practical experience in

that subject. You would not believe it, but there are only two or three good books, on the gas-producer and the use of gas-producers, which speak seriously of the use of producer gas in connection with the great steel industry and the great glass industry, which are the two most important users of producer gas. Therefore, when I was asked to get up a paper to be read before you, I found, on going to the libraries, that help was to be obtained mainly, not from the books which are labeled "producer gas," but from books which are devoted to the practical glass business and the practical steel business. There we frequently found something written by men who were using the stuff and knew what they were talking about.

Now coming to a short history of producer gas: In general, in a broad way, I should say that the producer-gas business in this country, in the first place, has been connected mainly with the use of soft coal. It has been a business which has produced gas in enormous volumes, and in a practically continuous system, and the system has been virtually "fool proof." There has been a tendency of the soft coal to coke on top. There is a tendency for the coal to clinker at the bottom. There is an inevitable production of tar. There is an inevitable production of soot. There must be a regular supply of air, and there must be a regular supply of steam. These seem to be the natural difficulties that the bituminous-coal producer was up against when it started. As far as we can obtain the history of the development of the gas-producer, we shall find that it has been compelled to pay attention to all these points at once.

Now in coming to consider the main point of the evening's work let us look at the fundamental chemical equation and the heat-energy equation involved in the *production* of producer gas. I shall deal this evening almost entirely in British thermal units, the English measure, and in cubic feet, etc., but I shall ask you to remember that, roughly, the calorie is almost equal to four B.t.u. The diagram shown in Fig. 1, will, I think, enable you to see what I mean. And I think it is very much better to take one or two equations and work them out carefully and try to understand them clearly.

When carbon oxidizes it goes to CO and then CO_2. If we study the heat relations between these two points we shall find, roughly, that of the *total* energy which is given out by C in burning to CO_2, about 30 per cent. is given up in passing to the CO state, and the other 70 per cent. in going to the CO_2 state. They tell us the best psychology is that which teaches us to draw the best mental pictures to assist our imagination. Therefore let us consider that the carbon is a stream of water pouring over a precipice, and going down to the CO level where it gives off 30 per cent. of its energy; then from the CO level it falls to the CO_2 level where it gives off the other 70 per cent. of its energy (Fig. 1).

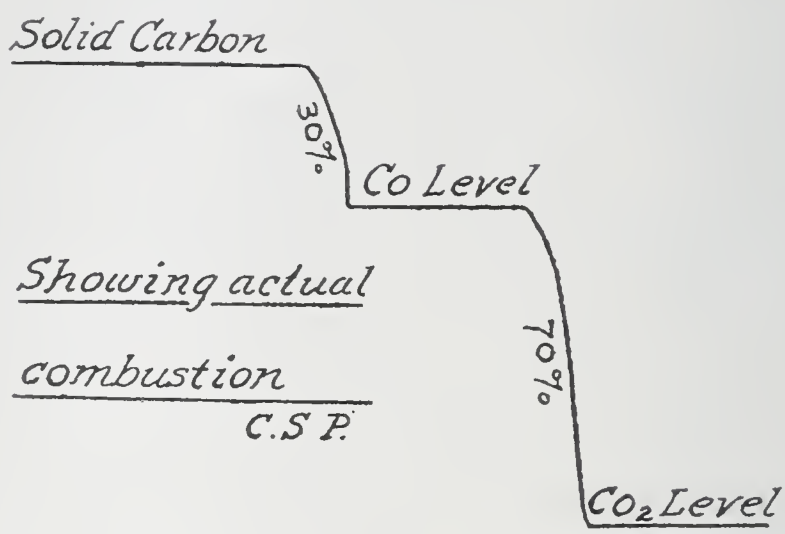

Fig. 1.

Right here I would call your attention to something very interesting. A great many students have examined the question and have asked "Why is it that the addition of one atom of oxygen to the carbon produces 30 per cent. of the energy, whereas the addition of a second atom of oxygen to the carbon produces 70 per cent.?" There should be something to explain the difference between the 30 and 70 per cent. It would look as though the addition of the first atom of O, to the C, ought to produce as much energy as the addition of the second atom of O. You will find that several of the authorities—for example Molinari, the Italian chemist—explain it in this way; that the carbon is not free carbon. What is carbon? What are we talking of, solid or liquid or gaseous carbon? Gaseous carbon we know almost nothing about. We know very little about liquid carbon. They

tell us that in a certain part of the arc there must be a little liquid carbon but we do not see it. Solid carbon we know very little about. It is probable that carbon is not "C," but "C_n" or "C_x" in charcoal or coke. There is an unknown number of atoms of carbon in the molecule and it takes a great energy to pull them apart. So the molecule of solid carbon is probably a group of atoms very strongly united and that missing "40 per cent." of energy is probably used in tearing apart the carbon atoms. This is not proved yet, but I ask you to notice this. In the correct picture you will have to think that before the carbon can fall to the CO level it must climb a hill which represents the 40 per cent. deficiency taken up in tearing apart the atoms of carbon (Fig. 2).

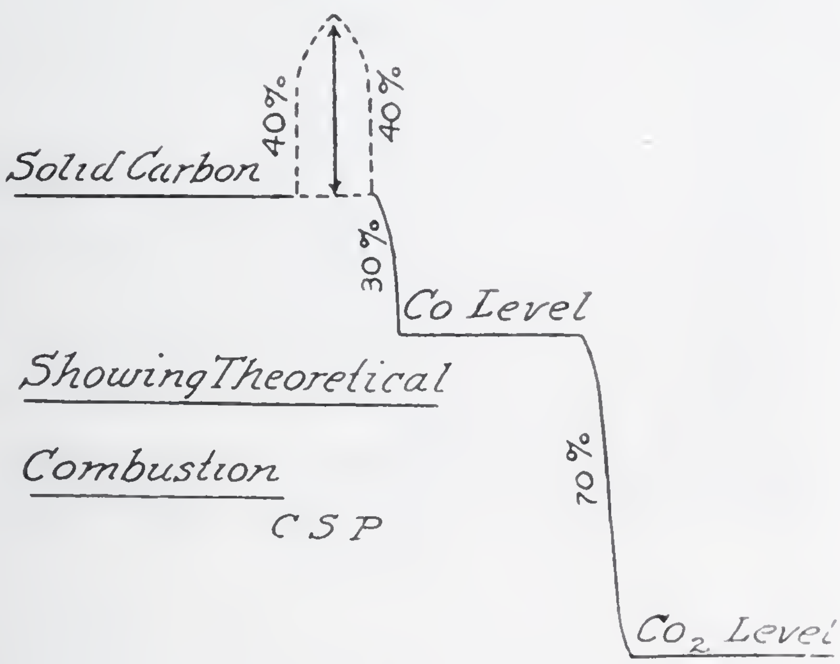

Fig. 2.

Now let us write out the two reactions—the so-called old producer-gas reaction and the water-gas reaction. When carbon burns, it goes to CO. Probably what happens is that it first goes to CO_2, and then as this CO_2 passes up through hot layers of carbon, the average is dragged back to CO; and this reaction which we shall discuss in diagram later on, is one of the most significant reactions in chemical dynamics. We should remember this—that in general, in handling carbon with oxygen, either with or without an excess of carbon, above 800 degrees C. it tends to go to the CO condition; and below 800 degrees C. it tends to go to the CO_2 condition (Fig. 3).

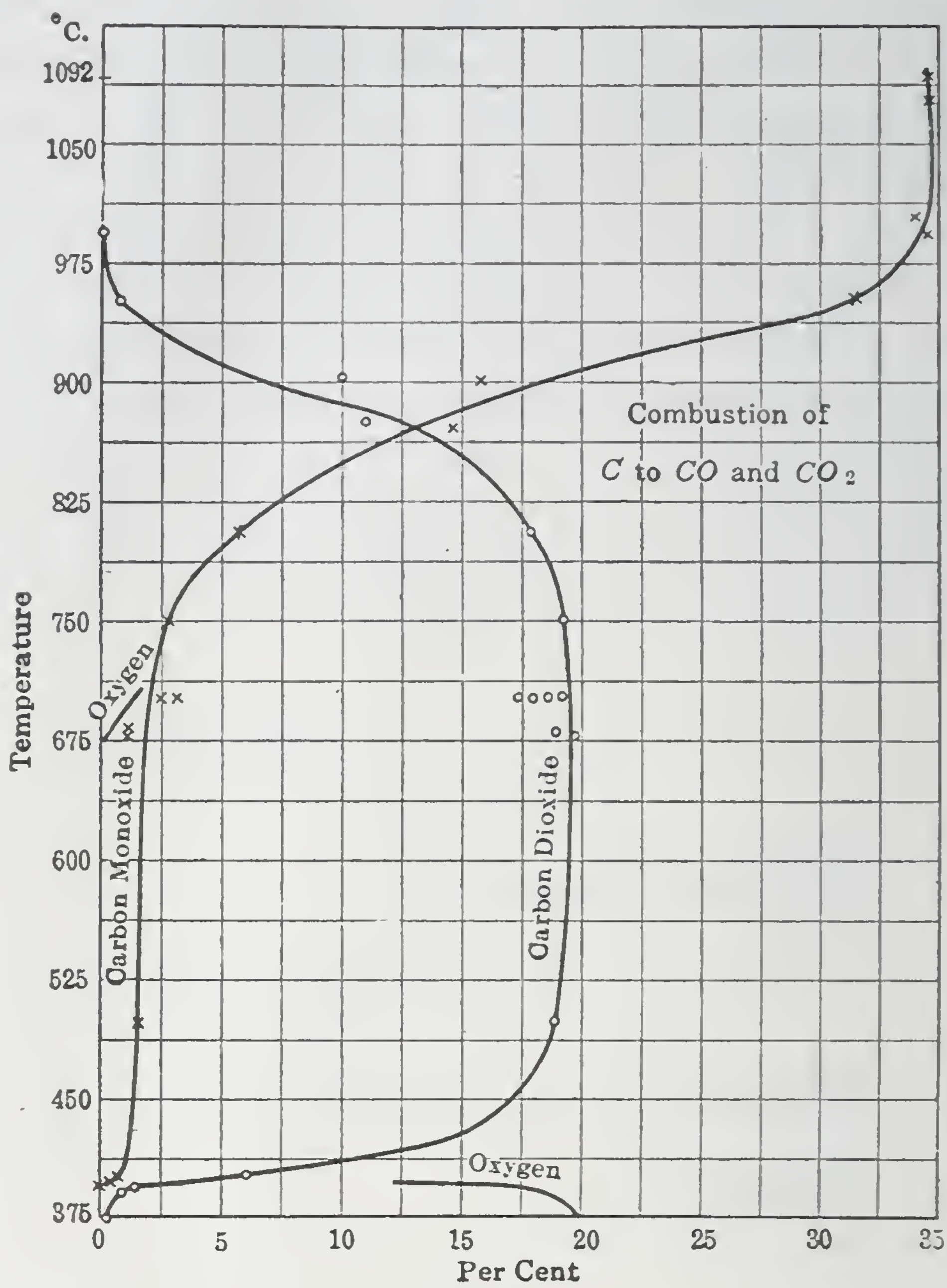

Fig. 3. Curves of Combustion of Carbon to CO and CO².*
Note the crossing of the curves at about 850° C.

Unfortunately a great deal of the magnificent research work which has been done can not be applied directly to our engineering work because instead of simple carbon and oxygen, we use steam and all kinds of hydrocarbons and all the volatile matter of soft coal, and the reaction becomes very much more complicated than the books tell us.

Next, the water-gas reaction. In the producer-gas reaction the result largely obtained is that one atom of carbon burns to CO and gives off 30 per cent. of its energy. But that carbon burning in the producer heats a neighboring atom of carbon. And, for the time, we speak as though we had coal to deal with. If

*From H. O. Hofman's "General Metallurgy," 1913. p. 295.

that hot coal is hit by steam, we get CO and H_2. Tie these two reactions together and we get, from two atoms of C, two volumes of CO and one volume of H_2; so that in general, in the practical producer-gas reaction, we should have roughly two volumes of CO and one volume of H_2, and that is practically what we do get in ordinary producer gas.

But that is not the only thing. We can change coal into volatile fuel at a cost of 15 per cent. of its energy. That is what producer gas means. It means that 85 per cent. of its energy would be left for burning. Practically the best work I have ever seen was a producer that gave about 82 per cent. efficiency. Of course there are many heat losses to explain this.

Theoretically, the first carbon atom, in burning to CO, took only one-half molecule of oxygen or one-half atmosphere of air, and for one atmosphere of air we could get twice that combustible gas. Therefore, for every molecule of oxygen used, we get two of the CO molecules in each reaction. If there is 20 per cent. oxygen in the air, that means that we are going to get 40 per cent. of CO here. Add to that the H_2 and we get 20 per cent. more.

Now in the case of the 20 per cent. of oxygen in the atmosphere of air, we have seen that its place is taken by 40 per cent. of CO, plus 20 per cent. of H_2—or 60 per cent. of combustible gas. Add this 60 per cent. of combustible gas to the 80 per cent. of residual nitrogen in the one atmosphere of original air, and we have 140 per cent., as a new basis on which to reckon our final percentage; and the 40 per cent. of CO, the 20 per cent. of H_2, and the 80 per cent. of nitrogen—when figured back to the 140 basis, as the new "100 per cent."—give us respectively:

CO	28.57%
H_2	14.29%
N_2	57.14%
Total	100.00%

Thus, theoretically, we should get, from hard coal or coke, a producer gas, with less than 60 per cent. of nitrogen and with over 40 per cent. of combustibles—CO and H_2. Practically, we do get, usually, a producer gas with over 60 per cent. of nitrogen

and with less than 40 per cent. of combustibles. The following table is valuable as showing the actual heat losses as figured by H. H. Campbell* from actual practice at Steelton, Pa., though it should be mentioned that the fuel used was soft coal, which ought to give a slightly richer gas than hard coal:

Lost as carbon in ash.........	2.1%
Sensible heat of dry gas......	13.7%
Sensible heat of steam in gas..	0.7%
Radiation and conduction (by difference)..........	5.1%
Total	21.6%

This loss of 21 per cent., as compared with the theoretical 15 per cent., is perhaps about the average, though I think I have known of practice averaging about 18 per cent.

So much for the production of producer gas. How about the burning of it? When we get a gas, we rate its heat value on the amount of heat units that one cubic foot will give off, and we rate the cubic foot in B.t.u. A B.t.u. is the heat required to raise one pound of water one degree F. in temperature. If natural gas is said to have a heat capacity of 1000 B.t.u. that means that one cubic foot of natural gas, burned under ideal conditions, would raise 1000 pounds of water one degree F. in temperature. In an ordinary coal gas of 500 or 600 B.t.u. that means that a cubic foot of coal gas would raise 500 or 600 pounds of water one degree F. in temperature. Producer gas is a very lean gas. It has a B.t.u. value from 100 to 140—sometimes, but rarely, as high as 160 or 170. That means that a cubic foot of producer gas in burning will heat 140 pounds of water one degree F. in temperature. We all know how our natural-gas supply is gradually going. Yet producer gas is taking its place and, strange to say, though natural gas carries five to eight times as much heat as producer gas, we are getting about the same heat in our mills and factories. We are making the same steel, we are making the same glass; indeed our furnaces would not stand a much hotter

*Manufacture and Properties of Iron and Steel. Ed. 2, 1903, p. 229

fire than we are getting. This remarkable fact, which is well known to engineers in this field, did not come to me in its full emphasis until a short time ago—that the so-called lean and poor gas gives almost exactly as hot a flame in burning as a rich gas. You would hardly believe that was so. In other words, producer gas gives practically as hot a flame *in burning* as natural gas. How can that be? Let us see.

How much oxygen is in the air? Approximately 20 per cent. The rest is nitrogen. The richer the gas is, the more oxygen we must have to make it burn; and the more oxygen we get to make it burn, the more of that 80 per cent. of nitrogen we have to mix with it. So when we get our actual mixture, the heating powers of the actual mixtures for the various gases vary only slightly.

To illustrate this principle of the relative heats produced by the various gases in burning, I have calculated a short table, which will show the figures for producer gas, for natural gas, and for coal gas. The respective compositions were assumed as follows:

Producer gas	%	Natural gas	%	Coal gas	%
CO	23.0	CH_4	95.00	CO	6.00
H_2	10.0	H_2	1.00	H_2	49.00
CO_2	7.0	CO	1.00	CH_4	35.00
N_2	60.0	CO_2	1.00	CO_2	2.00
		O_2	1.00	N_2	2.00
		N_2	1.00	C_2H_6	6.00

The heats of combustion of the various ingredients were taken as follows:

CO	343 B.t.u.
H_2	344 "
CH_4	1048 "
C_2H_6	1870 "

The respective weights for 1000 cubic feet, and for one cubic foot, for some of the ingredients of the common gases are as follows:

	1000 cu. ft.	1 cu. ft.
H_2	5.56 lb.	0.00556 lb.
O_2	88.96 "	0.08896 "
CO	77.84 "	0.07784 "
N_2	77.84 "	0.07784 "
CO_2	122.32 "	0.12232 "
Argon	111.20 "	0.11120 "
Air	80.51 "	0.08051 "
CH_4	44.48 "	0.04448 "
C_2H_6	83.40 "	0.08340 "

Now, in calculating the relative volumes of the respective burning gas mixtures, it should be noted that each of the respective combustible ingredient gases requires theoretically the following volumes of oxygen:

H_2	½	vol. of O_2 in burning to H_2O
CO	½	" " " " " " CO_2
CH_4	2	" " " " " " CO_2 and H_2O
C_2H_6	3½	" " " " " " CO_2 and H_2O

Thus CH_4 takes *two whole* volumes of oxygen *as such;* but as one volume of atmosphere carries only about 20 or 21 per cent. of oxygen, it would take *ten* volumes of air to burn CH_4, assuming that all the oxygen is used up and that there is no residual oxygen left in the flue-gas, but as some indications point to the value of some little residual oxygen in the flue-gas, I have assumed this residual oxygen at five or six per cent.; which gives us the following table of the relative amounts of air and combustible gas:

Heat-volume table of combustion mixtures of gases and air.

	Producer gas	Natural gas	Coal gas
Gas to air by volume	1:1	1:12 2/3	1:6½

But, as the gases are not all combustible but carry some neutral "filling", we find next:

Table of percentage of combustible in burning mixture, considering both the combustible and the supporter of combustion.

Producer-gas mixture	Natural-gas mixture	Coal-gas mixture
24.75%	20.8%	25%

Now the total heat produced by each of the respective combustible gas mixtures, herein considered comparatively, is as follows:

	Producer gas	Natural gas	Coal gas
Heat from the combustible gas mixture with air, in B.t.u.	11 339.0	99 560.0	55 594.0

And dividing each of these by its respective number of cubic feet in the respective *burning gas mixture,* gives us:

	Producer gas	Natural gas	Coal gas
B.t.u. from 1 cu. ft.	56.7	72.8	74

This is the remarkable result to which we have been moving in the last few paragraphs of figuring—that all of the heats produced per cubic foot of combustible mixture are of the same order, and very near each other. This is the explanation for the well known fact that the great glass- and steel-works of this wonderful region, are still running on, and using what is apparently low-grade producer gas, with about the same efficiency and economy that they developed with the old unlimited supply of natural gas. The figures are quite probably only approximate struggles towards the real facts; but, as reliable comparative figures, they are presumably explanatory and instructive. My own attention to their importance in the general fuel problem, was not definitely aroused till quite recently, though, I am quite sure that they are well known to the best experts in the leading industries concerned. And it is still more remarkable that I have been able to find no such statement of the fundamental facts of

the relative heats produced per cubic foot of combustible mixtures in burning, in any of the published material available.

These facts must be of fundamental importance, and they are submitted to the chemical engineering profession, with the hope that they may serve to stimulate further study along these lines.

Of course, this style of calculation can be carried much further; but, while I shall not be able to follow it through, I will briefly indicate some of the lines towards which our attention is to be directed in the immediate future.

There is the question of the relative advantages to be gained from the time-honored practice of preheating the gas and the air—one or both. If they are preheated, of course they carry on an immense quantity of heat that is otherwise "waste" and lost. On the other hand, in this very preheating there is bound to be a thinning out both of combustible gas and burning air—which means dividing the weight of the gas burned and the air for burning, by *two* or *three,* and perhaps more, depending on the degree of preheating. It must be realized what this means—to thin out our reacting gases two or three times, or more. Probably the *relative* thinning out of the three gases—producer, natural, and coal gas—will come out about the same, judging from the relative results obtained thus far, both in theory and practice. And it must be remembered that there is a new producer now on the market—the Smith—which uses the principle of cooling the gas before burning, with no preheating; and this form of practice is now being used by some of the large and progressive plants.

Following my earlier line of thought a bit further, here are some comparative figures of the actual weights per cubic foot of the respective gases, and their *burning mixtures* as used above:

Producer gas	Natural gas	Coal gas
0.073725 lb.	0.045981 lb.	0.03197 lb.

The relative weights of the three gases, with the respective proportion of air required for burning each, and in the actual burning mixture, are as follows:

	Producer gas	Natural gas	Coal gas
Gas to air by volume	1:1	1:12 2/3	1:6½
Weight of this gas-air mixture per cu. ft.	0.077117 lb.	0.074619 lb.	0.07404 lb.

Now these figures, referred to one pound of water, give, for each cubic foot of the actual burning mixture, the following:

	Producer gas	Natural gas	Coal gas
Ratio of each burning gas mixture by weight per cu. ft., to 1 lb. water.	1/13 or 10.077	1/13.5 or 10.075	1/13.5 or 10.074

Now if we figure on, to the condition *after* combustion, we note that in burning, by volume, the combustible gases, *with* the required oxygen, shape themselves much as follows:

1 vol. of CO + ½ vol. of O_2, give 1 vol. CO_2
1 vol. of H_2 + ½ vol. of O_2, give 1 vol. H_2O
1 vol. of CH_4 + 2 vol. of O_2, give 1 vol. CO_2 + 2 vol. H_2O
1 vol. C_2H_6 + 3½ vol. O_2, give 2 vol. CO_2 + 3 vol. H_2

Thus it appears that there is some contraction in the case of the burning of the CO and O_2; some in that of the H_2 and O_2; none in the case of the CH_4 and O_2; and actually some expansion in the case of the C_2H_6 and O_2. On the whole, the combustible parts, counting both the combustible and the supporter of combustion, do not show much tendency to alter the *total* volume of the whole burning mixture; for it must be remembered that most of the volume is occupied by the inactive nitrogen of the air.

Again, taking the actual heats in B.t.u. produced by each of the respective mixtures, per cubic foot, we had as follows:

	Producer gas	Natural gas	Coal gas
Heat produced per cu. ft. of burning mix.	56.7 B.t.u.	72.8 B.t.u.	74 B.t.u.
Weight of each mix. cu. ft. as comp. with water 1 lb.	1/13	1/13.5	1/13.5

Multiplying these last two sets of figures vertically, to get the real temperature per pound of each mixture in burning, and thus to find the heat rise produced in each mixture by its own burning—because the B.t.u. refers to the heat required to raise the temperature of one pound of water—we get as follows:

	Producer gas	Natural gas	Coal gas
Rise of temp. F.	737.1°	999.0°	1080.8°

But there are also the corrections for the relative specific heats of the gas mixtures as compared with that of water, weight for weight, and for the loss of heat in each mixture, due to the thinning out of the gases by expansion. These are contrary to each other, and may about compensate each other.

The specific heats of some of the principal gases concerned are as follows:

Relative specific heats of the gases, weight for weight, as compared with that of water which is 1

Water	1.000
Air	0.240
Argon	0.123
Nitrogen	0.250
Oxygen	0.230
Steam (superheated)	0.500
Hydrogen	3.4
CO	0.242
CO_2	0.200 to 0.270
CH_4	0.593
C_2H_6	Not given

Of course, most of these figures increase with rising temperature. Taking the rough average of the gas mixtures *after* burning it will be found that the specific heats are from one-quarter to one-half that of water; which means that the temperatures produced in the table which is given at the top of this page must be multiplied by some such figure as two, or three, or four. On the other hand, we must consider the heats from the thinning of the gas contents by expansion, that is the

expansion and thinning from the heat of the actual burning of the gases. But, if the gas and the air are both *preheated* in the checkerwork to use the waste heat from the flue-gases, that involves another set of complications, the end of which is not yet in sight. But the fact is that the use of what we regard as lean producer gas is able to furnish temperatures sufficient to do the work of such industries as those of glass and steel.

There is of course another fact that should be emphasized—namely, that while the rich gas, as coal gas or natural gas, must be diluted with air in order to get the amount of oxygen requisite to burn it, yet when so diluted it gives about the same quantity of heat per cubic foot of burning mixture. Now that does not mean that the rich gas has not more inherent heat capacity; for the lean gas, as producer gas, will not stand diluting with air to anywhere nearly the extent that natural gas or coal gas will. And that means that it is time to stop selling and buying gas by the cubic foot, and to buy it on a B.t.u. basis. The laws and the civic practice, both state and municipal, are moving towards that end; and all our influence should be devoted to putting the sale of gas on a rational basis. This is particularly true in view of reports that some companies are at present supplying the gas shortage by pumping in producer gas. Presumably the dilution has been necessary, and it is also probable that it is largely in the way of tentative experiment; but the attention of engineers should be called thereto to help safeguard the public interest. Private manufacturers can presumably be trusted to insist on paying for the actual B.t.u. in the gas served them. We are quite liable to see some curious accidents and some dangerous ones in the near future in the working out of the question of substitutes for natural gas; and for large municipal consumption, with long transportation, it goes without saying that some gas richer than low-grade producer gas will have to be found, however well the latter may work for the large local plant.

A few weeks ago I had the pleasure of meeting a gentleman who very kindly told me the history of the gas-producer in connection with the glass business. I had hoped to have some help from a friend in the steel business, as I can not be said to know anything practically about the use of gas-producers in that

industry, having visited steel plants only eight or ten times. Speaking of the glass business my friend, Mr. Loomis, of the H. H. Dixon Co., told me he has been in the business 25 years and the gas-producer was an old story at the time he began; how old he did not know, but he thought it had been developing 30 or 40 years before that. Siemens in 1850 or 1860 seems to have been the originator. It is said that for centuries the Welsh used a producer for making CO, though they did not call it a "producer." We know the Welsh are very secretive in their metallurgical processes. At any rate it was the Siemens producer that came to this country and seems to have been the first type of producer. The regenerator was used, also, for heating air and gases, and that seems to have come to this country at least 40 years ago. There were many imitations, and so-called improvements on the Siemens producer. These were all of the "static" type—that is, the tendency of the coal to coke and clinker had to be overcome by *hand* stirring. Some of this type of producer are still being used. Then there came in the "mechanical" type with rotating bottom, and automatic stirring on top. The Wood producer is one of these.

In the history of the gas-producer there were certain types developed to meet certain needs which afterwards disappeared. Several of these are the Nicholson, the Murphy, and the Gill. These came at a time when there was a great deal of cheap slack coal to be obtained and, as that special condition passed away, the use of the special producers was discontinued; but I presume that some producers are successfully handling slack to-day. There are now two standard types, the static and the mechanical.

Of the producer troubles, I have mentioned tar and soot. I did not realize how great a difficulty the soot was and with what practical success inventors and engineers have met the difficulty. It seems that wherever possible every bituminous plant is now arranged with a by-pass, so that, in glass-works, for instance, about once a week the draft is shut off and the by-pass turned on and by letting the air in it burns itself out. It is said the accumulation of soot in a flue will be from one to two feet thick all the way around, and the flue will be almost plugged up. A good many producers try to write their reaction as though they

got along without steam; they make their air moist or have a water-seal or water pan. But almost all the producers now swing the gas through by the million feet; and they actually handle 20 to 24 tons of coal in 24 hours, each ton of coal producing from 100 000 to 130 000 feet of gas, so each producer is producing several million cubic feet in 24 hours. Anything swinging along that way must have power behind it and naturally there would be an air blast and a steam blast. The pressure carried is all the way from a few inches of water to 12 or 15 pounds of steam or air. Some of the glass authorities recommend 15 to 20 pounds pressure.

Another thing. The producer should always be built or designed for its maximum production. No producer can run at half capacity and produce the best results. If it is built to consume 24 tons of coal in 24 hours, it should not be compelled to slow down and run only five or ten tons in that time, or it will not get good results. Troubles will certainly arise.

I wish to thank you and to say that in securing data I am greatly indebted to Mr. Loomis of the Dixon Company, Mr. McClelland of the Carnegie Library of Pittsburgh, and your Secretary, who have assisted me in producing a great deal of this paper.

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DISCUSSION

Mr. E. B. Stimpson:* I think of a case of a producer that Dr. Palmer did not show. They do not use a stirring device at all in the producer, they rake the coal backward and forward over the top and do not tear any holes in the fire. They claim a very high percentage of CO on account of this fact. They also claim, from a series of tests running over a period of five months on this same producer, a B.t.u. of about 185. I refer to a Morgan producer of the latest type. I would like to ask if Dr. Palmer considers such claims are reasonable from a scientific and theoretical point of view.

Dr. C. S. Palmer: I think any B.t.u. over 120 or 130 is almost always explained by a glance at the gas analysis. Quite a number of the producers are able to do what all ought to do to a greater extent—to save a large part of the volatile matter in the coal. The volatile matter is very largely methane and this methane has a B.t.u. of 1000. There is one type which I merely mentioned this evening, and that is the Smith producer, which advocates the idea that we can get more gas to burn if we do not heat it, but keep it cool, because the gas expands and therefore dilutes itself on heating. With this producer a B.t.u. value of 170 is reported, together with the ability to run the coal so as to get about three or four per cent. of methane, and every per cent. of methane means three times the heat power of CO or H_2. Ordinary producer gas carries one per cent. or less of methane but in the analysis I did not mention that because it was not worth considering. It is an important point, however, if we want to get the richer gas. Remember, I stated that a richer gas had no advantage over other gas.

There is another point I would like to bring up. When I visited the open-hearth steel plant I noticed that with the producer gas where the air and gas came in over the ports the air came in above and the gas below, and the gas was in a yellowish cloud indicating that there was a great deal of suspended matter

*Manager, F. J. McWade Co., Pittsburgh.

in it. There are two points of interest here. I had a friend in the chemical department of a plant at Braddock a few years ago, whose work was to get a sample of the producer gas every day. Now, in the shape in which it is burned, it is in a yellowish smoky state, but in the shape in which he analyzed it, it was a colorless gas, so I suspect his analysis of producer gas very rarely showed the actual state of the case. The gas that goes into the furnace is probably loaded with vapors. Those vapors do two things. A large part of that smoke may be steam but a large part of it is tar and that tar is a fuel and I should not be a bit surprised, if we could get at that cloudy gas and analyze it, to find that the actual B.t.u. would run from 150 to 200.

Another point. When that gas burns, it burns with a luminous flame. Now it seems to be a belief of the older steel men around here that the presence of illuminants in any burning gas causes that gas to radiate heat better. If you were to burn an absolutely colorless producer gas mixed with ordinary hot air that would produce an altogether invisible flame and there would be very little body in it to radiate the heat down. In an open-hearth furnace the heat has to be radiated down through five or six inches of slag into the steel that is being treated, and the practical men of the middle-age school tell me that illuminants undoubtedly increase the effective radiation of the burning gas. In talking with one or two of the younger engineers around here I find they are inclined to doubt that. Many of you have seen some of the surface-combustion furnaces that radiate heat almost around corners, and a perfectly colorless burning atmosphere with no illuminants whatever except what come from the sides of the hot brick. It is generally believed, however, that the presence of illuminants in the gas does increase the radiation in the open-hearth furnace.

MR. C. G. GERBER:* In transferring producer gas through large-diameter mains is there any change effected in the heating quality of the gas due to loss of heat from radiation?

*Engineer, Crucible Steel Co. of America, Pittsburgh.

Dr. C. S. Palmer: It is a guess and I do not know that I could guess any more intelligently that you could. I should say producer gas is used very acceptably the way it is now used by transmission short distances, 50 to 200 feet. Where it is being used for public service purposes—and I suppose there are a dozen or more of us who know that producer gas is being shipped into this city—the people who ship it in take their chances of having their pipes clog unless they clean the gas. If they clean it I do not see why it should not behave pretty well. It is a pretty heavy gas—about the molecular weight of 25—and is 55 to 65 per cent. producer gas, with a molecular weight of 28, while the molecular weight of marsh gas is 16. You will have just as hot a burning gas at the port because it will not take as much air as the richer gas. But it would not be fair for the company to charge as much for that gas as natural gas, though I do not see any objection to putting producer gas into any sort of supply if it is sold on a B.t.u. basis.

The danger of carbon monoxid poisoning should be kept in mind. Carbon monoxid was always present in the old anthracite practice, and I dread to see any large-scale introduction of producer gas, with its high content of carbon monoxid, here in the East where nine out of ten cities use a water-gas for public service purposes. In city gas, which is essentially a water-gas, the raw water-gas contains from 35 to 40 per cent. of carbon monoxid and from 45 to 50 per cent. of hydrogen and has no illuminating value except when burned in a Welsbach burner.

For years the gas companies have been obliged to meet a false standard and carburet the gas. Five or six years ago it did not cost the big manufacturers and public utilities more than eight to twelve cents a thousand to make raw water-gas, but the addition of oil to bring it up to the candle-power standard raised the cost to 22 to 28 cents by the time the gas reached the gasometer. Thus it cost more to carburet the gas than to make it. but the oil used was the lowest grade obtainable and the vilest smelling oil the Lord ever made. In one respect this was fortunate. Carbon monoxid is odorless and hydrogen almost so and the gas would be infinitely more dangerous if its presence were

not advertised by the impurity which is in the shape of the oil used to carburet it.

If the law is changed and we are permitted to use raw water-gas with a B.t.u. content of 350 it will be a good gas to ship around anywhere, but it will not have a bad odor and it will be a very dangerous thing. Its use in kitchens should perhaps be restricted and it should certainly never be permitted in any sleeping room. In the case of gas companies using producer gas or water-gas it may be necessary to have a legal requirement, similar to that in some German cities where the addition of a bad-smelling oil is compulsory—not to increase the candle-power, but to give an odor that can be recognized.

As for the transferring of producer gas through pipes I do not see any objection except that it is a more sluggish gas and there will be more friction and it must be well cleansed and you have to look out for CO.

MR. C. G. GERBER: In asking my question I had in mind long mains in the steel plants, and what effect there would be, if any, upon the quality of the gas, due to cooling.

DR. C. S. PALMER: It would not have any effect if you get it out of the producer. I see your point. There might be a short period there where it would cool down and the chances are it would cool down rather quickly. Of course it is possible for the reaction between C, CO and CO_2 to take place without any great amount of free carbon around and there might be a reaction. Offhand, however, I do not see any objection to it.

MR. JOHN C. CARR:* In a producer using about 20 tons of coal in 24 hours, what is the proper thickness of the fuel bed above the zone of combustion?

DR. C. S. PALMER: Burning soft coal? I could not answer that. If I could get into overalls for a week I could answer, but I have not had the practice. And that is one reason why I am

*Superintendent, Track Department, Jones & Laughlin Steel Co., Pittsburgh.

appearing under false colors in assuming to talk on a subject of which I know almost nothing on the practical side. But I should say that in general, just as a guess, there would be perhaps two feet; from what I have seen in other bituminous producers.

MR. JOHN C. CARR: It runs about 18 inches with us.

MR. A. STUCKI:* I was in hope of getting a real lively discussion on this important subject. Every winter we are now compelled to look for substitutes for natural gas. Oil, pulverized coal and producer gas are often chosen, and it looks as if the latter could be applied with the least change in the original plant. There are many here to-night who have had a great deal of experience with producers and we would like to hear from them; and perhaps our President sanctions the plan of receiving written discussion.

DR. C. S. PALMER: The hydrocarbons are wanted in the furnace. We do not want to waste them. Pittsburgh coal will run about 35 per cent. volatile, most of which is hydrocarbons. If we could isolate them we would have very similar characteristics in heat values to natural gas. And we want that gas along with the gas which comes from the partial combustion of the fixed carbon in the coal. But the difficulty is that when some fresh charge of coal is dumped into a hot producer a large evolution of hydrocarbons results immediately and they go over in a volume and pass through the heating furnace, before all the rich carbon constituents can be completely burned, and pass out of the stack only partially burned. We do not want to burn these gases until they burn in the secondary combustion which takes place in the furnace where the producer gas is being used.

MR. R. L. HANAU:† The author has pointed out very clearly the quantitative and qualitative requirements which are essentials of producer-gas manufacture. Being in intimate touch

*Consulting Engineer, Pittsburgh.

†Consulting Engineer, Bacharach Industrial Instrument Co., Pittsburgh.

with the instrument line, it naturally suggests itself to me to inquire whether the producer-gas manufacturer, in general, employs all useful aids in that direction. I believe he does not. You, as specialists, know the reasons why in many cases you prefer to be guided by experience, though it often leads to very unsatisfactory results. In all modern manufacture it is necessary to have quantitative and qualitative records, however, not for the shelves of the department but for serious study for the advancement of the art.

I brought this subject before you because I believe it within the range of the paper of to-night. We were very clearly shown what has to be done and how we have to proceed, and many difficulties, etc., were pointed out to us. Much of the data brought before us will be of limited value to the progressive manufacturer of producer gas, unless he is equipped with accurate and convenient means to measure and record the operating conditions of his plant.

Mr. J. C. Hobbs:* Dr. Palmer, I believe, has devoted practically the entire evening to the discussion of single-zone producers. It would be greatly appreciated if he would give us some comments on the double-zone type of producer. It seems to me that if one of the great troubles in handling high-volatile coal is the lack of uniformity of the gas, due to the rapid volatilization after each fresh charge is put in, then it would be logical to use that type of producer which best overcomes these difficulties. I mention this point in order to obtain the information because my own experience has been along the single-zone type.

I am very much interested in the chemical equations and in Dr. Palmer's explanation of the great difference in the amount of heat of combustion of carbon to CO, and to CO_2. The water analogy was also very good. There is still, however, one question in my mind, and that is: Why is it, in the burning of carbon in natural gas, we still get the same amount of heat from the carbon to CO reaction that we would when burning pure carbon to CO? Is this explained by the fact that the carbon appears in

*Assistant to Superintendent of Power Stations, Duquesne Light Co., Pittsburgh.

combination with hydrogen, and that a part of the energy is used to break the bonds between the carbon and the hydrogen?

I notice in the chemical equations that Dr. Palmer went first from C to CO and then to CO_2, whereas the curves indicated that the first reaction was from C to CO_2, and then later, if the temperature was sufficiently high, the secondary reaction of CO_2 to CO took place. This order of reactions was well presented in *Technical paper* 137 of the United States Bureau of Mines. "Combustion in the Fuel Bed of Hand-Fired Furnaces," by Kreisinger, Ovitz and Augustine. In this very interesting paper it was pointed out that only 6½ pounds of air could be forced through a uniform incandescent fuel bed, and that the additional air to change the CO back to CO_2 had to be supplied above the fuel bed.

Dr. C. S. Palmer: At one point I tried to correct that, where I said that in all ordinary furnaces probably the first combustion was to CO_2 and then afterwards to CO.

I have had no practical experience with the actual working of single- and double-zone producers. I know that the design is to produce more uniform gas, and it has been done in some cases. But my guess is that the double-zone producers are, in practice, not fool-proof. I have never seen any in operation where they did produce good uniform results with ordinary workmen as they are supposed to.

Mr. F. L. Egan:* The reasons for presenting anything from my limited experience with gas producers are:

1. The conditions were unusual at that time and perhaps would be, even to-day.

2. The gas from the same battery of producers was used in all the heating processes of a large plate-glass plant and also burned directly in three Koerting single-cylinder, two-cycle gas-engines, of 600 horsepower each, which had formerly used Indiana natural gas quite successfully.

We started the plant, using coal from three different mines as we could secure it. An average proximate analysis of these coals was:

*Engineer on River Equipment, Carnegie Steel Co., Pittsburgh.

Moisture 12.14 per cent., volatile 35.17 per cent., fixed carbon 43.73 per cent., ash 8.96 per cent., sulphur 3.54 per cent., B.t.u. dry coal, 13 000.

This coal was used in two static-type producers of a well-known make, rated at 20 tons of coal per 24 hours. These producers were handled by an operator who knew nothing of the theory of gas making but who had about 12 years of practical experience.

The gas was passed through a mechanical tar extractor of the fan type—in which the gas is whirled around in a spray of water and the tar and water are drawn off through a sealed water leg—and also through a tower washer or scrubber of the hurdle type. The gas was then carried approximately 800 feet through an overhead main of steel plate to the three engines before mentioned and, from the same main, branches were taken to the various heating furnaces at an average distance of somewhere around 400 feet from the producers.

The producer gas gradually replaced the natural gas in all heating furnaces around the glass plant, even to a casting-table equipped with an automatically controlled heating system, all quite successfully.

In the meantime we had increased the producer battery to six and then eight producers by contracting the steelwork to local boiler shops and making the patterns and castings for the necessary cast parts. We were never able to distinguish any difference in results between our first two and the later producers which we made and installed.

Here is the point I wish to bring out: This plant was making a success of supplying small plants—in this case furnaces—from a central producer plant, thus replacing natural gas; but as a power-producing plant burning producer gas direct in gas-engines it was almost a complete failure. The gas was dirty when it reached the engines, and in passing through the long intricate passages and the piston-valve gas pump of the Koerting engine the tar and lampblack accumulated and adhered until we were obliged to dismantle the engine and clean it out with various curved chisel bars, etc. While the engines ran with load enough to keep them hot we had little trouble, but stopping for two hours

usually meant a broken valve-stem or rocker arm and delay before getting on the line again. On three occasions I had to hitch a chain-block to the gas-pump piston-valve and even then had difficulty in pulling the valve out, due to tar and soot which had cooled around the valve.

The gas value at the engines was so irregular that we usually had to shift part of our load to the steam plant two or three times during the 24 hours and at times as much as 40 per cent. would be so shifted.

At the same time the furnaces operated entirely satisfactorily, proving that the dependable production of power from producer gas and gas-engines is difficult unless the gas is cleaned to a very low fraction of dust and other solid impurities, or the engines are designed to handle dirty gas.

There was in this particular case no way of accurately checking power costs as we did not meter our gas. However, I have compared the cost of maintenance and all fixed and labor costs, exclusive of the gas we used, against the total power costs per kilowatt-hour of a 2000-horsepower steam plant which I was looking after at the same time and in which we burned the same coal, and in which we had similar labor and operating conditions. The gas-plant power costs, exclusive of gas burned, equalled 74 per cent. of the total cost of power in the steam plant.

This is presented not as a comparison of steam and gas power, but to bring out the great difference in the two following problems:

1. Replacing natural with producer gas in heating furnaces in metallurgical and glass processes.

2. Replacing either the coal-fired steam plant or the gas-engine plant burning natural gas, with the gas-engine plant burning producer gas.

In closing I wish to add that, in this state (Pennsylvania), we had a mechanic employed regularly on the test block, testing gas engines, who collected fairly large damages for a permanent injury to his health due to being rather frequently overcome with gas.

Mr. B. Kramer:* While the producer-gas manufacturer aims at gasifying 85 per cent. of his coal which, as Dr. Palmer showed us, was theoretically possible, would it not be worth his while to consider that one of the sources of his troubles and losses, especially with bituminous coal, is the loss of the by-products. By first carbonizing the coal either in by-product coke-ovens or in bench retorts—the latter being preferable when no blast-furnace coke is wanted—the producer-gas manufacturer could convert close to 25 per cent. of his coal into gas, while the residual coke could be treated with air and steam in the ordinary way to make producer gas.

Part of the producer gas could be used for heating the retorts, and the coal gas could be treated for benzol and toluol in the regular way. At the present time when our Government demand for toluol so greatly exceeds the supply, this circumstance should be worthy of special consideration.

This is by no means a new idea, but as it was neither mentioned by the lecturer nor brought out in the oral discussion that followed, I thought it worth while to take the privilege of mentioning it.

*H. Koppers Co., Pittsburgh.

BROAD RAILS VERSUS DEEP RAILS*
WHEEL CONTACTS ON RAILHEADS†

CORRESPONDENCE‡

ATLANTIC COAST LINE RAILROAD COMPANY
Office of Chief Engineer.

Wilmington, N. C., Dec. 21st, 1917.

J. E. Willoughby, Chief Engineer.

Mr. G. H. Barbour,
6057 Stanton Avenue,
Pittsburgh, Pa.

Dear Sir:

I thank you for the blue prints you sent me of the paper that you are to read at the coming meeting of the American Association for the Advancement of Science. I have read the paper with much interest and while I have no desire to make a criticism of the paper, I would point out to you that in the third paragraph on page 2, tenth line, the statement, "85-lb. to 100-lb. rails are found to be inadequate," is not in accord with my experience. The A. C. L. R. R. is operating its main line with success with 85-lb. rails. We are of the opinion that the 85-lb. rail is adequate for the traffic which goes over our lines. We are able to handle any loads which our connections to the north can give us. We accept all loads up to Cooper's E-60.

The fourth paragraph, fourth line, I would suggest to be re-written to read,

> "al forces. Double fastenings are being provided on many rail-ways by the use of tie plates, which are punched for four spikes."

It is needless to say that I am not in accord with the theories of Dr. James E. Howard. The experiments which the Joint Committee on Stresses in Track of the A. S. C. E. and A. R. E. A.

*Paper by George H. Barbour. "Proceedings," July 1917, p. 327.

†Paper by George H. Barbour. "Proceedings," November 1917, p. 789.

‡[These letters are published as submitted to the Society, disregarding slight departures from the style of the "Proceedings."—Ed.]

are making will in my judgment prove that Dr. Howard's views are in the main in error.

Yours truly,

J. E. WILLOUGHBY.

6057 Stanton Avenue, Pittsburgh, Pa.

January 26th, 1918.

Atlantic Coast Line Railroad Company,

J. E. Willoughby, Chief Engineer,

Wilmington, N. C.

Dear Sir:

I have been pondering the following statement in your letter of December 21, 1917, ever since its receipt,

> "The A. C. L. is operating its main line with success with 85-lb. rail. We are of the opinion that the 85-lb. rail is adequate for the traffic which goes over our lines. We are able to handle . . . all loads up to Cooper's E-60."

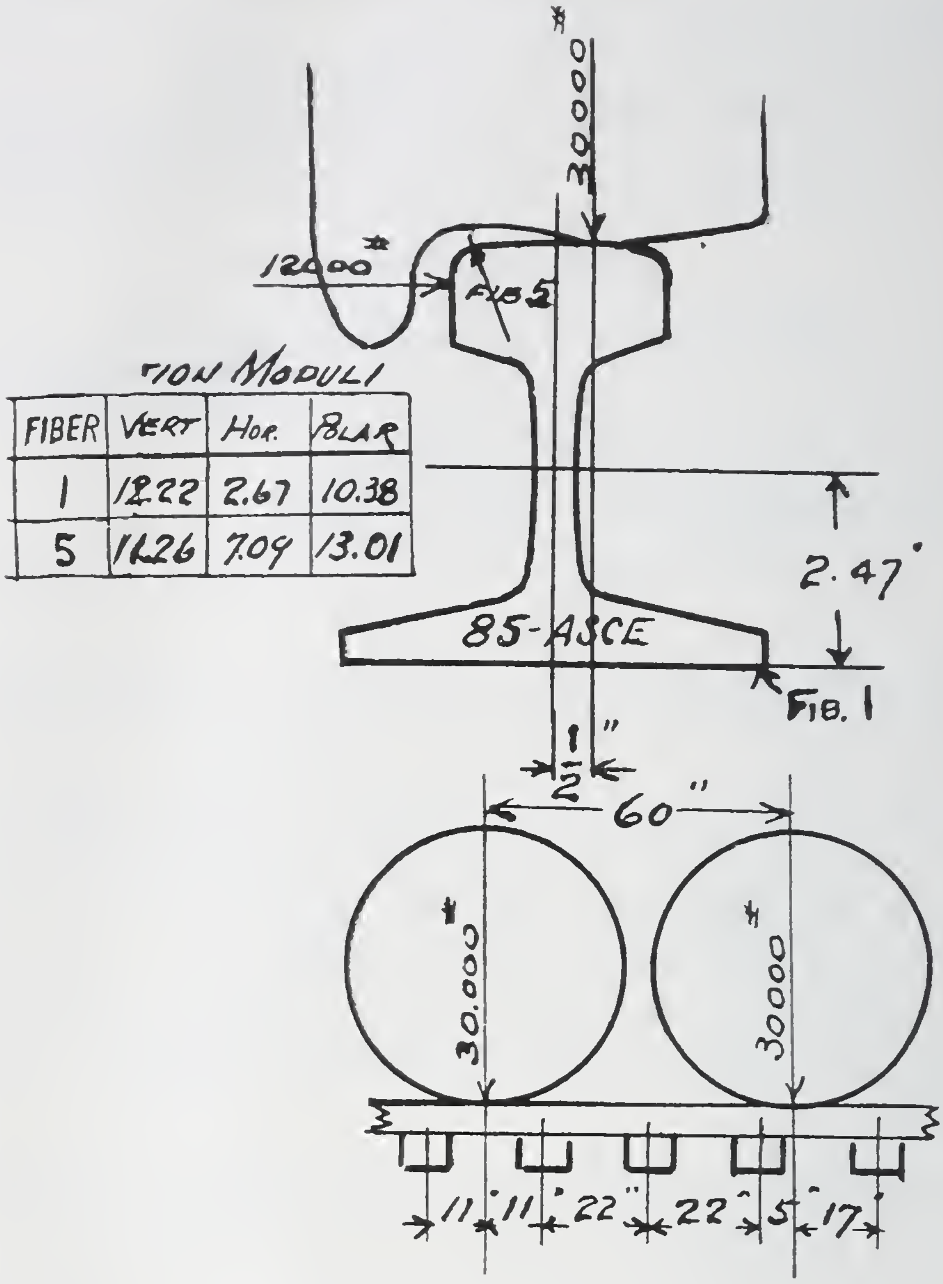
for other steamroads are doing the same thing, for which success the careful supervision of such experienced engineers as yourself is responsible.

Inspired by your statement, I have carefully investigated the critical fibers in flange and head of an 85-lb. A. S. C. E. rail under Cooper's E-60 loading, on $7 \times 9'' \times 8' \ 6''$ wooden crossties, 22″ centers, and double spiked at $3\frac{1}{2}''$ centers, as follows:

There is distributed by one rail to each crosstie under the drivers, $\frac{22 \times 30\,000}{60} = 11\,000$ lbs., and the maximum vertical bending moment at a wheel contact when halfway between a pair of crossties is

$$\frac{11\,000}{60^2}(11 \times 49^2 + 33 \times 27^2 + 55 \times 5^2) = 158\,409 \text{ in. lbs.},$$

but with worn wheel treads the load might be applied outside the vertical axis of the rail, producing a torsional or twisting moment, which for $\frac{1}{2}''$ eccentricity would amount to $30\,000 \times 0.5 = 15\,000$ in. lbs., of which but half would affect the fibers of any given cross-section of the rail. By the formula for flexure and torsion combined,



$$T = \sqrt{\left(\frac{Mp}{Sp}\right)^2 + \left(\frac{Mb}{2Sb}\right)^2} + \frac{Mb}{2Sb},$$

where T = combined stress in fiber investigated,

Mp = twisting moment, or $\frac{15\,000}{2}$ in this instance,

Sp = polar section modulus for the fiber investigated,

Mb = bending moment, or 158 409 in this instance, and

Sb = section modulus for the fiber investigated; we obtain for fiber 1, the critical fiber of the flange,

T, vertical =

$$\sqrt{\left(\frac{15\,000}{2 \times 10.38}\right)^2 + \left(\frac{158\,409}{2 \times 12.22}\right)^2} + \frac{158\,409}{2 \times 12.22}$$

$$= 13\,005 \text{ in. lbs.},$$

showing that the twisting due to the eccentric application of the wheel load has but slight effect.

The rail also may be subjected to lateral thrusts aggregating 20% of the whole axle load on account of the wheels being single flanged; 20% of 60 000 = 12 000 lbs., producing a horizontal bending moment of $\frac{12\,000\ (22—3.5)}{8} = 27\,750$ in. lbs. This horizontal force being applied ½″ below the top of the rail produces a twisting moment of 12 000 times the vertical distance from the point of application to the center of gravity of the rail, or 12 000 (2.72—0.5) = 26 640 in. lbs. Hence in fiber 1,

T, horizontal =

$$\sqrt{\left(\frac{26\,640}{2 \times 10.38}\right)^2 + \left(\frac{27\,750}{2 \times 2.67}\right)^2} + \frac{27\,750}{2 \times 2.67}$$
$$= 10\,550 \text{ in. lbs.}$$

In addition there are the stresses induced in the fibers by the curves of the transitory depression. As closely approximating results obtained by Dr. Dudley, I assume the depression under the locomotive drivers to be in the ratio of 1″ for each 100 lbs. per sq. in. pressure on the bearing area of the crosstie, in this instance, $\frac{22\,000}{9 \times 102 \times 100} = 0.24''$, and that half of this, or 0.12″, represents the deflection for the 240″ span influenced by the four drivers of the E-60 loading. By the deflection formula for a span uniformly loaded, with one end fixed and the other end supported, which this curve closely approximates,

$$\text{T, depression} = \frac{0.12 \times 23.15 \times 29\,000\,000y}{240^2} = 1399y,$$

where y = the vertical distance from the rail's center of gravity to the fiber in question, or for fiber 1,

$$\text{Td} = 1399 \times 2.47 = 3456 \text{ in. lbs.}$$

By summation we obtain the following combined stresses for fiber 1,

a. for the condition of no lateral thrust,

13 005 + 3456 = 16 461 inch lbs

4 115 for impact.

b, for condition of 12 000 lbs. lateral thrust on one rail,

$13\,005 + 3456 + 10\,550 = 27\,011$

$6\,753$ for impact,

33 764

c, for the condition of winter heavings and hard and soft spots, where there is an increase of 3456 in. lbs. for each ¼″ increment in the depth of the transitory depression. Of course where this condition is known, the dangerous spot is traversed with care and with the minimum of lateral thrust, without which an 85-lb. rail would experience,

$1.25\,(13\,005 + 4 \times 3456) =$ 33 536

equivalent to 1″ depression before reaching the stress of the previous condition..

Similarly, fiber 5, the critical one of the railhead, receives from the vertical loading a stress of

$$Tv = \sqrt{\left(\frac{15\,000}{2 \times 13.01}\right)^2 + \left(\frac{158\,409}{2 \times 11.26}\right)^2} + \frac{158\,409}{2 \times 11.26} = 14\,091$$

and from the lateral thrusts,

$$Th = \sqrt{\left(\frac{26\,640}{2 \times 13.01}\right)^2 + \left(\frac{27\,750}{2 \times 7.09}\right)^2} + \frac{27\,750}{2 \times 7.09} = 4166$$

and induced by the transitory depression,

$Td = 1399 \times 2.67 =$ 3735

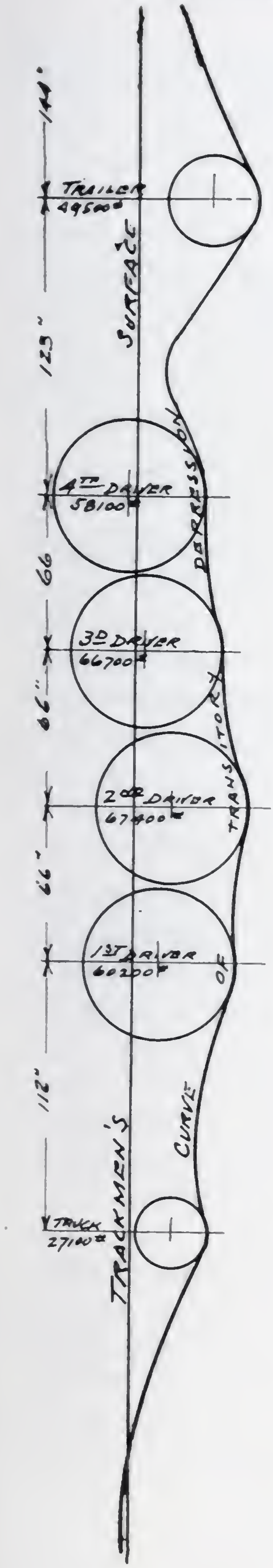
Combining,

a, = ... 22 283
b, = ... 27 490
c, = ... 36 289

Dealing with a material of close to 50 000 lbs. elastic limit, you can figure for yourself your factor of safety under the various conditions.

Yours very truly,

G. H. Barbour.



6057 Stanton Avenue, Pittsburgh, Pa.,
February 12, 1918.

Hon. W. G. McAdoo, Director General,
United States Railroads,
Washington, D. C.

Sir:

As the result of extended investigations of steamroad track, the writer has arrived at certain conclusions which he takes the liberty of laying before you; and, though the time might seem inappropriate, the necessity of bringing our railroads to their maximum efficiency should the war continue, and of restoring them fully equipped for all prospective requirements whenever it ends, must appeal to you, who already have experienced their inadequacy, after their decade of repression, in trackage, in equipment, in terminal facilities, and even in capacity to earn their own replacements and betterments, to meet the exigencies of ordinary boom times, let alone the stern urgency of wartime emergencies.

Steam railroading is based on the distinctively American principle of a flexible superstructure upon an elastic roadbed, whereby the dynamic energy of a self-impelling train is dissipated by many wheel contacts through the bending rails, rebounding crossties and compressible ballast, accompanied by a depression of the portion of track immediately under the wheels. This subsidence, from which the track returns to the trackmen's surface or normal level directly the train has passed, very aptly has been named the transitory depression; and one is illustrated on the left-hand margin with the deflections greatly exaggerated. The head of the rail is in compression at the

wheel contacts and in tension between them with its base vice versa, hence repeated alternate vertical deflections occur as the wheels traverse their attendant transitory depression. Though this cushioning action of the track is the salvation of railroading, excessive deflections of the rail are the great defect of track; for, as no conceivable rail has sufficient strength to more than follow the vertical curves of the transitory depression, its fibers are bound to experience the stresses induced by those curves in addition to the stresses resulting from the bending and torsional effects of the wheels' vertical loads and lateral thrusts; and steel is sure to rupture after a certain number of repetitions of excessive stresses, though able to stand more repetitions of the smaller magnitude that would result from diminished deflections.

Tremendous concentrations of pressure on insignificant areas of contact between wheeltread and railhead have a cold-rolling or peening effect on the exterior fibers of the latter, inducing such tension in adjacent interior fibers as to cause their rupture after repeated excessive deflections. These defects, known as transverse fissures, are the more insidious in that their presence is not suspected until the rail breaks. All qualities of steel are subject to this defect, which, however, may be ameliorated by decreasing deflections.

The joint is the great defect in rails, due to 14 inches, more or less, of the railend being held so rigidly in the vise-like grip of the splicebars that the undulatory continuity of the rail is interrupted at each joint, with the result that the rails soon acquire permanent sets at the splices, producing a dangerous and rough-riding track. Improvement lies in decreased deflections. Worn and ragged wheels, improper loading and spacing of wheels and neglect of maintenance are among the abuses; and all the above-mentioned features will be found thoroughly discussed in the exhibits forwarded you under separate cover, comprising: July and November PROCEEDINGS, Engineers' Society of Western Pennsylvania; paper on "Railroad Track" read before the Dec. 28, 1917, meeting of the Association for the Advancement of Science, and copy of letter, dated Jan. 26, 1917, to J. E. Willoughby, Chief Engineer, Atlantic Coast Line, which exhibits a more correct method of solving this intricate problem than was

employed before the Engineers' Society, where I had hoped for discussion on this point, but was disappointed.

Yours very respectfully,

G. H. Barbour.

ATLANTIC COAST LINE RAILROAD COMPANY
Office of Chief Engineer.

Wilmington, N. C., Feb. 15th, 1918.

J. E. Willoughby, Chief Engineer.

Mr. G. H. Barbour,
6057 Stanton Avenue,
Pittsburgh, Pa.

Dear Sir:

I beg to acknowledge receipt of your letter of January 26th.

The results obtained therein are not altogether in accord with the tests which have been made by the Committee of the American Society of Civil Engineers for determining stress in track. A progress report of the Committee was made to the Society at its January meeting. If you do not get the publications of the American Society, kindly advise me and I will send you a copy of the report, since I think it will be of interest to you.

In this connection I beg to thank you for a copy of the Proceedings of the Engineers' Society of Western Pennsylvania.

Yours truly,

J. E. Willoughby, *Chief Engineer.*

DIRECTOR GENERAL OF RAILROADS
Interstate Commerce Building.

Washington, February 18, 1918.

G. H. Barbour, Esq.,
6057 Stanton Avenue,
Pittsburgh, Pa.

Dear Mr. Barbour:

The Director General requests me to acknowledge your letter

of February 12th and to thank you very much for the suggestions therein contained.

You will of course appreciate that vast numbers of suggestions bearing upon railroad problems have been made and intelligent consideration of all of them cannot be had immediately. At the earliest possible moment, however, these matters will receive careful attention.

Cordially yours,
Oscar A. Price, *Private Secretary.*

6057 Stanton Avenue, Pittsburgh, Pa.
March 2, 1918.

Atlantic Coast Line Railroad Company,
J. E. Willoughby, Chief Engineer,
Wilmington, N. C.

Dear Sir:

Am just in receipt of your letter of the 27th ultimo with copy of January, 1918, *Proceedings* of the American Society of Civil Engineers, containing "Progress Report of the Special Committee to Report on Stresses in Railroad Track," for which I thank you. This is the first opportunity I have had to realize the extent of its activities, though, of course, I knew such a research was in progress. What I have gathered from skimming over its pages seems confirmatory of my findings; and as soon as the matter has been fully digested I will be heard from in extenso.

I note that 0.083 Pl to 0.088 Pl is offered as an approximation of the vertical bending moment, which for Cooper's E-60 loading would give, $0.083 \times 30\,000 \times 60 = 149\,400$ to $0.088 \times 30\,000 \times 60 = 158\,400$ inch pounds, against the 158 409 inch pounds obtained by my formula.

Am still obsessed by the pertinent point raised in your letter of December 21, 1917:

> "The A. C. L. is operating its main line with success with 85-lb. rail. We are of the opinion that the 85-lb. rail is adequate for the traffic which goes over our lines. We are able to handle . . . all loads up to Cooper's E-60."

when everybody knows that such roads as the P. R. R. cannot duplicate your performance.

According to the Pocket List of Railroad Officials for Second Quarter, 1917, the above-mentioned roads compare as follows:

	A. C. L.		P. R. R.	
Miles of track	4 775		5 388	
No. of locomotives	832		3 994	
No. of cars	30 875		173 465	
Locos per mile of track	$\frac{832}{4775}$	$= 0.174$	$\frac{3994}{5388}$	$= 0.741$
Cars per mile of track	$\frac{30\,875}{4775}$	$= 6.05$	$\frac{173\,465}{5388}$	$= 32.19$

showing that the P. R. R. has $\frac{0.741}{0.174} = 4.26$ times as many locomotives and $\frac{32.19}{6.05} = 5.32$ times as many cars per mile of track as the A. C. L., which would indicate, were all other conditions equal, *four or five times the number of rail undulations and amount of wear in a given period of time.*

Cordially yours,

G. H. Barbour.

6057 Stanton Avenue, Pittsburgh, Pa.

March 9, 1918.

Hon. William G. McAdoo,
Director General of Railroads,
Washington, D. C.

Dear Sir:

Supplementing mine of the 2nd instant, please note accompanying copy of letter of even date to J. E. Willoughby, Chief Engineer, A. C. L., regarding Progress Report of Rail Stress Committee, published in January, 1918, *Proceedings* of the American Society of Civil Engineers.

It is unfortunate and disappointing that the bulk of these important tests were conducted on a stretch of track not up to the standards of the principal American railroads. Crossties 8′ 0″ long on 6″ to 12″ of ballast constitute insufficient roadbed for modern traffic. Many foreign roads use 9′ 0″ ties, while our fathers bequeathed us 8′ 6″ crossties on 12″ to 18″ of ballast, and

instead of lopping anything we should be adding on. The Committee found greater stresses in the outer than in the inner flange of the rail, considerable of which resulted from the rail inclination caused by the flexure of the short crosstie, which complicates the separation of the stresses produced by the lateral forces from those caused by the eccentricity of the wheel contacts.

Again, these are fair weather tests, with the stage set, the parts sandpapered, the counterweights adjusted and even the sun excluded; without consideration of those critical factors, vital and variable, that cover the wear and tear of track and equipment, the impetus and imperfections of mechanisms, the tolerances and limitations of materials and manufacture, the mutations of the seasons, the fury of the elements and the frailties of human nature; all of which I have endeavored to compensate in my recommendations. In fact, the Committee seems to have laid more stress on the minutiae of precision that on broad and general principles of engineering and the best traditions of American steamroad practice.

Very respectfully,

G. H. BARBOUR.

6057 Stanton Avenue, Pittsburgh, Pa.
March 9, 1918.

Atlantic Coast Line Railroad Company,
J. E. Willoughby, Chief Engineer,
Wilmington, N. C.

Dear Sir:

Continuing discussion of the Progress Report of the Rail Stress Committee, last mentioned in my letter of the 2nd instant, in line with Dr. Dudley's diagrams of tests conducted on the N. Y. C., my calculations have been based upon a transitory depression subsidence in the ratio of 1″ to 100 lbs. per sq. in. pressure transmitted through the bases of the crossties. This gave me 0.24″ subsidence under the drivers of the E-60 locomotive. The Mikado illustrated on page 135 of the Progress Report with drivers spaced 66″ and averaging 27 000 lbs. produced an average subsidence of 0.27″ in 24″ of ballast, which would indicate a subsidence ratio of 1″ to 87 lbs. per sq. in.; for $\frac{54\,000}{3 \times 8 \times 96 \times 87}$

$= 0.27$, with $6'' \times 8'' \times 8'\ 0''$ crossties, 22″ centers. This would make the depression factor for the 264″ span influenced by the drivers, $\frac{0.135 \times 23.15 \times 30\,000\,000y}{264^2} = 1346y$; and the vertical bending moment, $0.088 \times 27\,000 \times 66 = 156\,834$ inch pounds. As I understand the Stress Diagrams to illustrate the mean of the stresses at the two edges of the base of the rail, and the effects of the lateral forces to appear under the heading, "Ratio of the Stresses at One Edge of the Base of Rail to the Mean of the Stresses at the Two Edges," the lateral thrusts may be neglected in comparing my results with theirs, so my formula becomes, $T = 1.25 \left(\frac{156\,834}{22.9} + 1346 \times 3\right) = 13\,609$, against the 13 000 obtained by the Committee in the case of the 125-lb. rail on 24″ of ballast illustrated on page 186, Figure 94, 1916 tests at 35 miles per hour.

Similarly, for the D. L. & W. Mikado, Figure 102, page 192, 105-lb. rail on $7'' \times 9'' \times 8'\ 6''$ crossties and 18″ of ballast, at 40 miles per hour,

$$\text{Depression} = \frac{237\,000}{12.4 \times 9 \times 102 \times 87} = 0.24'',$$

$$\text{Depression factor} = \frac{0.12 \times 23.15 \times 30\,000\,000y}{272^2} = 1126y,$$

$$\text{Vertical bending moment} = 0.088 \times 29\,625 \times 68 = 177\,276,$$

$$T = 1.25 \left(\frac{177\,276}{17.2} + 1126 \times 2.77\right) = 16\,783,$$ against the 17 000 obtained by the Committee.

I do not understand how the bulk of these tests came to be conducted on a stretch of track that is anything but representative of the best American steamroad practice. 8′ 0″ crossties are too short and are responsible for considerable of the turning-out of the rail; for, with the wheel contacts at least ½″ outside the axes of the rails, the span of the rail reactions on the crossties is all of 60″, giving by the well-known formulae, for 33 000-lb. wheel loads on 8′ 0″ crossties, 22″ centers, bending moments of,

under the rail, $\frac{22\,000 \times 18^2}{2 \times 96} = 37\,125$,

and, at center, $\frac{22\,000\ (96 - 72)}{8} = 66\,000$, and no won-

der there was inclination of the rail, even with the locomotive at rest. For 8′ 6″ crossties the moments are about equal,

under the rail, $\dfrac{22\,000 \times 21^2}{2 \times 102} = 45\,500$,

and, at center, $\dfrac{22\,000\ (102 - 84)}{8} = 44\,000$, and when the increase of the moment at center due to the lateral forces is considered, a further increase in length seems desirable.

Thus it may be seen that the problem is capable of reasonable solution by old and familiar Anglo-Saxon methods of computation and without recourse to abstruse formulae, especially of German origin, covering several pages of the *Proceedings*.

Very cordially yours,

G. H. Barbour.

EFFICIENCY OF THE SCREW

Benjamin F. Groat*

Some fifteen years ago the writer computed tables and constructed diagrams for the purpose of making quick calculations of the performance of screws. A paper presenting the tables and diagrams, with illustrations of their uses, was read before the Section of Mechanical Science of the American Association for the Advancement of Science at the Chicago Convention, which assembled between December 30, 1907, and January 4, 1908. The tables and diagrams were not published, but have proved of such service to the author that he is now constrained to offer them for publication.

There is a tendency for practical men to avoid mathematics and mathematical formulas, to underestimate their value, and to try to substitute something "simple" in their place. The motive is undoubtedly good, and all have sympathy with efforts to simplify calculations and reduce the labor expended upon estimates; but in practical work, and in teaching, there is one fact which no amount of subterfuge will modify. The intellectual concepts of mutually related physical phenomena cannot be made simpler than a certain minimum of complexity which depends upon the nature of the particular phenomena observed. All efforts of practitioners and pedagogues to get below this irreducible minimum will fail, and may prove far more costly than boldly facing a bristling bunch of scientific principles.

In many cases it is not the mathematics or the science of the problem which makes the trouble, so much as a failure on the part of the practitioner to consider all the circumstances of the problem, including in particular the mutual relations of the forces, motions and physical parts involved. Simple as the screw is in principle, its mechanical forces, motions and parts take on several different aspects which must be clearly distinguished before any correct analysis can be effected.

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To show how much in error one may become by neglecting to take cognizance of the nature of a simple problem, the following problem, taken bodily from a text-book, will be discussed by means of the tables and diagrams presented herein.

"*Example*:—What pull must be applied at B to just raise 5000 pounds hanging to the screw? The screw itself weighs 111 pounds and has a pitch of one inch. R = 24 inches; f = 0.2."

"Then

Weight of screw..................	111.0
Friction due to screw, 111 × 0.2.....	22.2
Weight lifted	5000.0
Friction due to weight, 5000 × 0.2...	1000.0
W = ...	6133.2

$$\text{And } F = \frac{6133.2 \times 1}{6.283 \times 24} = 40.7 \text{ pounds."}$$

In explanation of the data of the problem it may be stated that a figure given in the text-book referred to represents B as the circumference of the horizontal hand-wheel which turns about a vertical axis. The nut is rigidly attached to the weight (5000 pounds) to be lifted, which, together with the weight of the screw, is supported by a collar bearing under the hand-wheel. R is the radius of the hand-wheel, while f is a general friction factor. The formula for F is given on the same page and corresponds to our formula 1, below. 6.283 is a constant of the formula. If the writer understands the figure correctly, the nut does not support the weight of the screw, but the collar bearing does.

One of the objects of this discussion is to show that a general coefficient cannot be employed in this manner to ascertain even approximately the torque upon a screw necessary to lift a given load. Each design has its own coefficient, and it would be but little better than a guess to attribute any particular coefficient to a given screw.

In the example quoted—as demonstrated on page 396 of this paper—we may obtain vastly differing results if we assume differing dimensions in different cases for certain elements entirely neglected in the statement of the example. In one case the required pull on the hand-wheel may be 89 pounds, while in another it may be 143 pounds. So that, while the text-book employs a friction factor of only 0.2, leading to a pull of only 40.7 pounds on the hand-wheel, the correct solution may give results three or four times as large with a coefficient of friction half as great as the one employed in the example.

If we consider the thrust, or load, to increase with an increase in friction, a likely condition where machinery is operated by a screw, the uncertainty occasioned by using a general friction factor is even more pronounced. This has been well shown, incidentally, by R. Camerer in an article concerning the friction of the gate-operating mechanism of a turbine. (See *Zeitschrift des Vereines deutscher Ingenieure*, vol. 50, Dec. 15, 1906, page 2030.)

Those who desire to make correct calculations relating to screws will have no difficulty in using the tables and diagrams by following the appended directions without the necessity of referring to the text. The problems related to the spiral ratchet screw-driver and the screws discussed in Tables V, Va, VI, and VIa illustrate principles and the uses of the formulas, tables and diagrams.

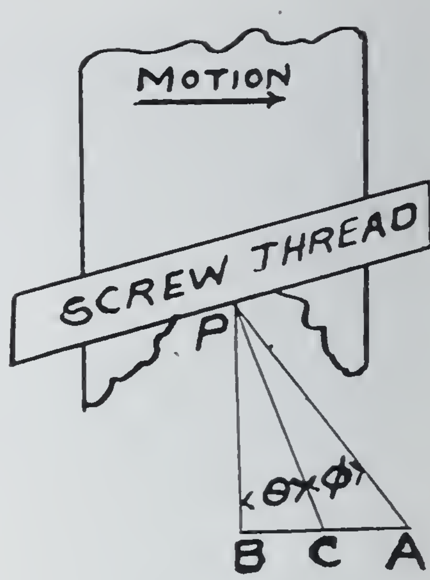
GENERAL CONSIDERATIONS

From the mechanical point of view, a screw is a machine which converts a torque into a thrust, or, in the contrary sense, converts a thrust into a torque. If we distinguish between work done in rotation and work done in translation a screw may be regarded as a transformer of torque into thrust, and vice versa.

A screw acts as an intermediary between three external bodies, one of which, however, may be considered to be a fulcrum and to be at rest. The mutual interactions between the other two bodies and the screw may then be considered under three distinguishable cases.

θ = angle BPC

ϕ = angle CPA



Case 1.

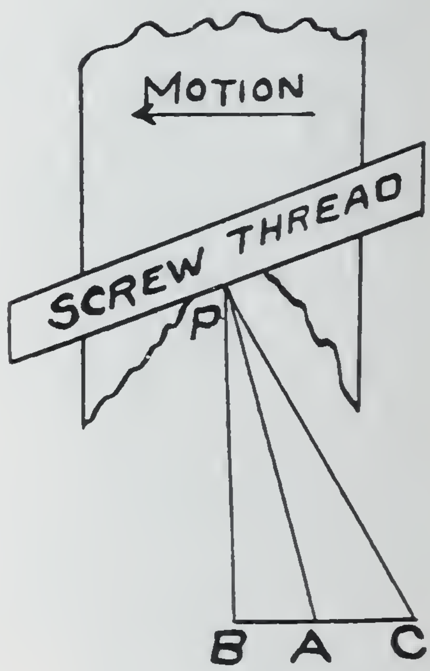
The screw receives a torque from one of the bodies and converts or transforms a portion of that torque into an equivalent thrust which it transmits to the other body. A jack-screw lifting a weight is an example of this case.

The figure is almost self explanatory if it be understood that the nut is fixed while the screw turns, and that θ (= angle BPC) and ϕ (= angle CPA) are, respectively, the angle of slope of threads and the angle of friction. The reasoning is the same if the screw is fixed and the nut turns, but the figure would require some corresponding modification. In this case the net reaction of the nut is shown by the resultant AP, of which a component, AB, opposes the motion.

θ = angle BPC

ϕ = angle CPA

$|\theta| > |\phi|$

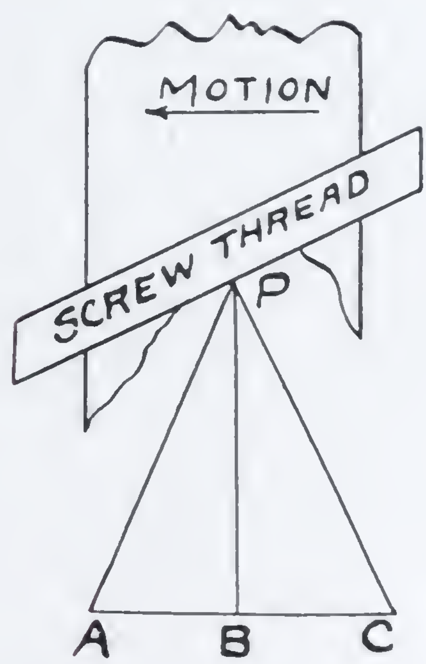


Case 2.

The screw receives a thrust from one of the bodies and converts or transforms a portion of that thrust into an equivalent torque which it transmits to the other body. A "spiral" ratchet screw-driver affords one of the few examples of this case. The slope of the screw-threads must exceed a certain minimum.

The figure is similar to that of case 1. However, the angle of slope, θ = angle BPC, is numerically greater than the angle of friction, ϕ = angle CPA, but of contrary sense. There is a component, AB, of the resultant, AP, which *assists* the motion. The motion *must* occur unless opposed by a torque upon the screw, of sense contrary to the impending motion and to the component AB. A limit of this case is reached when $\phi = -\theta$, that is, when A falls upon B, which implies that the screw locks and cannot turn unless assisted by an external torque.

$$\theta = \text{angle BPC}$$
$$\phi = \text{angle CPA}$$
$$|\phi| > |\theta|$$



Case 3.

The screw receives a torque from one of the bodies and a thrust from the other, the combined work of both the torque and the thrust being consumed in friction. A jack-screw letting a weight down is an example of this case. The slope of the screw-threads cannot exceed a certain maximum.

The point A in this case falls to the left of B, as shown in the figure. The motion cannot occur unless *assisted* by an external torque upon the screw of the same sense as the motion, but, as in all cases, contrary to the component AB, which, in fact, may be taken as a measure of the couple which must exist in the

screw spindle in order to maintain uniform rotation, remembering that bearing friction is an additional element not yet taken into consideration.

The mechanical elements necessary to effect these interactions are the screw spindle, the screw nut and the pivot or thrust-bearing. The thrust-bearing is dispensed with when the body concerned with torque receives the thrust and the torque is in the form of an axial couple, as in the case of the screw-driver and screw above, otherwise the thrust-bearing is necessary.

NOTATION AND REMARKS THEREON

The notation employed is as follows:

T = the thrust which is due to the reaction between the threads of the screw and the threads of the nut. It is not necessarily equal to the thrust delivered by the screw upon external bodies, considering the screw, nut and thrust-bearing to be a self-contained unit. This is because the weights of parts of the screw may affect the action.

t = the thrust upon the thrust-bearing. It is not necessarily equal to T though, where the weights of the affecting parts of the screw are negligible, it is usually so considered. The relative values of T and t thus depend in each case upon the particular positions of the screw and bodies acted upon.

M_t = the torque which is theoretically equivalent to the thrust, T, due to the reaction between the threads.

M = the actual torque passing between the screw or nut, as the case may be, and the body which receives or imparts torque.

M_s = the torque which exists in the body of the screw or of the nut, according as it is the screw or the nut which turns.

It is of importance to notice certain cases of relations which may exist between M_t, M and M_s. Suppose we take the case of the jack-screw lifting a weight. Neglecting the frictional resistance due to the side-thrust of the hand-lever upon the neck journal, the torque M is delivered by the lever to the screw or nut, as the case may be. A certain part of this torque is expended in overcoming the frictional resistance due to the thrust-bearing. The remaining torque, M_s, passes along the body of the turning element to the screw-threads where all but the effec-

tive torque, M_t, is expended in overcoming the frictional resistances of the threads. The effective torque is there converted into the thrust T. The mechanical efficiency of the jack-screw would, therefore, be $M_t \div M$.

In the case of the screw-driver, the thrust, T, equivalent to the torque M_t, would be exerted by the handle upon the screw. A certain portion of this torque would there be expended in overcoming thread friction, while the remaining torque, M_s, would pass into the body of the screw and be transmitted thereby directly to the screw to be driven. As there is in this case no thrust-bearing, the mechanical efficiency of the tool would be $M_s \div M_t$, neglecting advance of the driven screw.

When letting a weight down by means of a jack-screw, the torque M, neglecting side-thrust as before, and the torque M_t in the form of a thrust would both be delivered to the screw, and be entirely expended in overcoming the frictional resistances of the threads and thrust-bearing. Hence, by analogy, but without strict justification, we may consider either the ratio $M_t \div M$, or $M \div M_t$, to be the efficiency of the machine. This efficiency will be found to be negative when employing the general formula.

The efficiency of the threads would be $M_t \div M_s$, or $M_s \div M_t$, according as torque is being converted into thrust or thrust into torque.

The efficiency of the thrust-bearing, neglecting side-thrust, would be equal to $M_s \div M$, or $M \div M_s$, according as torque is being impressed upon, or delivered by, the screw. The latter case occurs when the threads have sufficient slope to permit the settling of the weight.

It will be of advantage to keep these two efficiencies separate, not only because there may be cases of pure thread friction, but also because thread and pivot frictions naturally separate themselves in mathematical treatment.

μ and ϕ are, respectively, the coefficient and angle of sliding friction for the threads. When thrust is being converted into torque (the weight descends) these quantities are both to be taken negative in the general formulas. "It appears that for screws lubricated with oil, with pressures of 1½ to 5 tons per square inch of projected area of screw thread, and at low speeds, the

coefficient is about 0.11 to 0.20 whether the screw or the nut are of steel, wrought iron, cast iron, or bronze."—Unwin. The quoted remarks are apparently based on the experiments by A. Kingsbury, published in the *Transactions* of the American Society of Mechanical Engineers, 1896, vol. 17, page 96, and also in the *American Machinist,* about Dec., 1895. These are by far the most extensive experiments that have been made upon screw friction. Since the most serious obstacle in applying the formulas, tables and diagrams of the present paper lies in determining correct values of the coefficients of friction, the results from Mr. Kingsbury's paper are summarized in the appendix.

μ_1 and ϕ_1 are, respectively, the coefficient and angle of friction for the thrust-bearing. They are probably somewhat less than the corresponding values for the threads. If the threads are not cut threads, they may be considerably less. They are both negative in the general formulas when the weight descends.

r = average radius of the screw-threads or arm of frictional resistance acting upon the surface of the threads. No great error will be made if it is taken as the arithmetical mean of the inner and outer radii of the screw-threads. If questions of elasticity are introduced, its exact determination becomes more difficult.

ρ = average radius (similar to that for r) of the pivot or collar.

n and θ are, respectively, the slope, and the angle of inclination of the threads.

p = the axial pitch of the screw.

GENERAL FORMULAS

With the foregoing notation the following general formulas concerning the square-threaded screw will be established. (See page 385.)

$$M_t = \frac{Tp}{2\pi} = T\ r\ n \qquad (1)$$

$$\frac{M_t}{M_s} = \frac{\tan\ \theta}{\tan\ (\theta + \phi)} = n\ \frac{1 - \mu n}{n + \mu} \qquad (2)$$

$$\frac{M_s}{M} = \frac{1}{1 + KA} \qquad (3)$$

Where $n = \tan\theta = p \div 2\pi r = p \div 6.283\ r$; and μ, μ_1, ϕ and ϕ_1 are to be taken negative when the weight is being let down or the screw slackened. Also in accordance with formula 3:

$$K = \frac{\tan\phi}{\tan(\phi + \theta)} \tag{4}$$

$$A = \frac{\mu_1}{\mu}\frac{t}{T}\frac{\rho}{r} \tag{5}$$

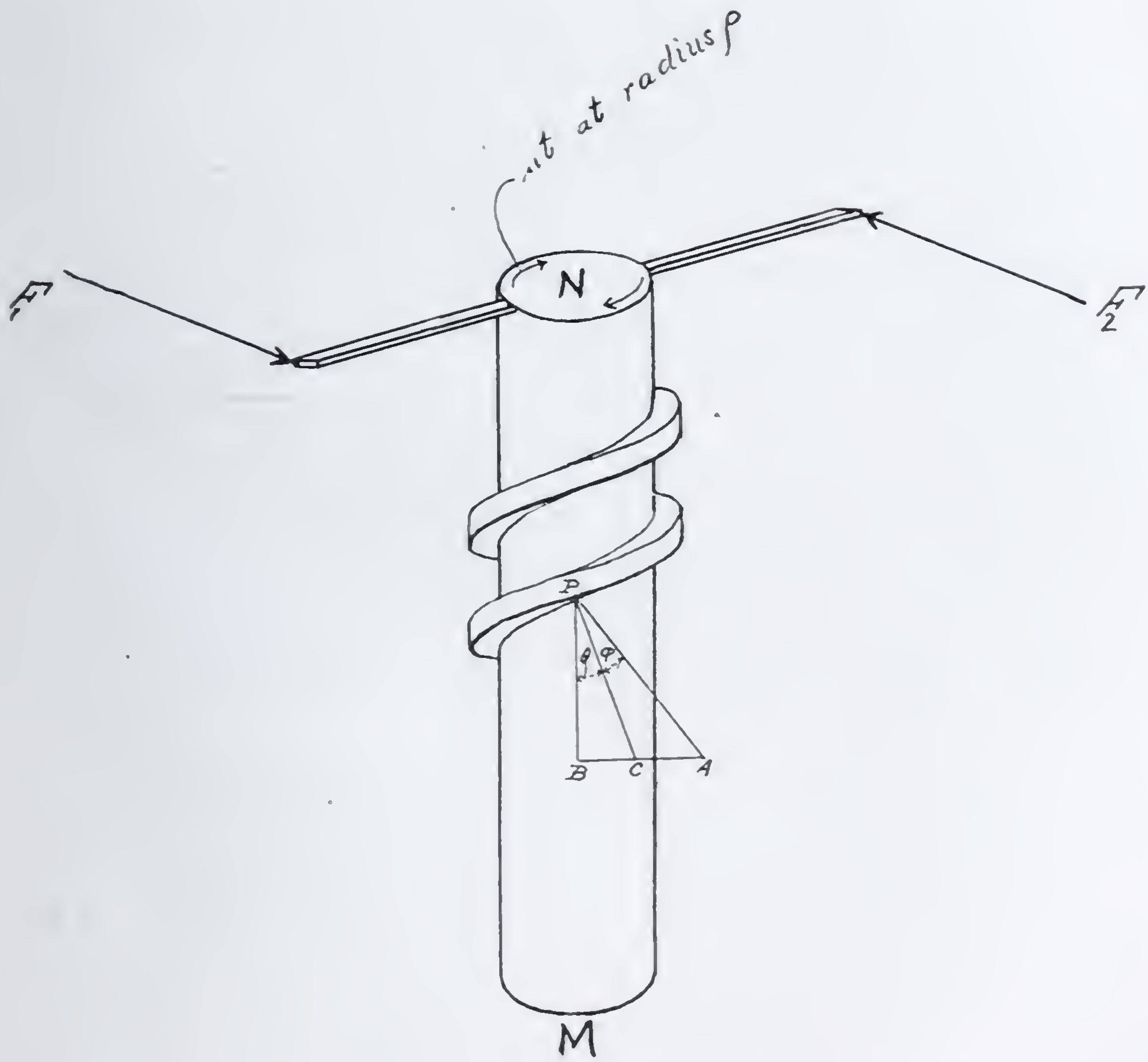


Fig. 1.

PROOF OF THE GENERAL FORMULAS

Formula 1 may be most easily proved by applying the law of the conservation of energy. That is, the work of the torque expended upon a frictionless screw is equal to the work of the thrust developed. Hence, employing the former notation,

$$\mathrm{Tp} = 2\pi \mathrm{M_t};$$

Therefore $$\mathrm{M_t} = \frac{\mathrm{Tp}}{2\pi} = \frac{\mathrm{Tp}}{6.283} \qquad (1)$$

If it is desired to find the force exerted upon a lever of length L, which will develop the torque M_t, it is necessary only to divide by L, thus:

$$\mathrm{F} = \frac{\mathrm{Tp}}{2\pi \mathrm{L}} = \frac{\mathrm{M_t}}{\mathrm{L}}$$

Let the couple of forces, F_1, F_2, Fig. 1, act through the equal lever arms upon the square-threaded screw MN. The direction of the couple is such that the rotating screw is thrust upward by the fixed nut against the thrust-bearing at N, thus tending to lift the load. It is required to find a formula for the efficiency of the threads and a formula for the efficiency of the thrust-bearing.

Consider an element of the thread at P. Let CP be a normal to the surface of the thread. Then if the screw were frictionless, the pressure of the nut upon the element P would be normal to the surface and might be represented by CP. Since the motion is supposed uniform, the screw is in equilibrium and the pressure, CP, of the nut would balance the element, PB, of the thrust, T, at P and the tangential element, BC, of the couple, M_t, at P. The same relation would hold for all the elements of the thread. Hence, θ being the angle of the threads,

$$\mathrm{BC}\ r = \mathrm{PB}\ r \tan\theta,$$

or $$\mathrm{M_t} = \mathrm{T}\ r \tan\theta.$$

In like manner, if there be friction, the reaction AP of the nut would be inclined at the angle $(\theta + \phi)$ with the axis of the screw, and we should have:

$$\mathrm{AB}\ r = \mathrm{PB}\ r \tan(\theta + \phi) \qquad (\mathrm{A})$$

or $$\mathrm{M_s} = \mathrm{T}\ r \tan(\theta + \phi) \qquad (\mathrm{B})$$

Therefore $$\frac{\mathrm{M_t}}{\mathrm{M_s}} = \frac{\tan\theta}{\tan(\theta + \phi)} \qquad (2)$$

If the screw rotates in the opposite direction to that supposed above, the angle ϕ becomes negative and should be so inserted in the formula. Hence, general formula 2 becomes established.

Turning now to the thrust-bearing, it will be seen that the friction upon the screw is contrary to the motion and must act upon some average lever arm, ρ. By the principles of friction, the frictional resistance is $\mu_1 t$, and its moment is $\mu_1 t \rho$ about the axis of the screw. If M is the torque impressed by the couple, then, since M must overcome all resistances,

$$M = M_s + \mu_1 t \rho$$

Hence $$\frac{M}{M_s} = 1 + \frac{\mu_1 t \rho}{M_s}\text{, dividing by } M_s.$$

Therefore $$\frac{M}{M_s} = 1 + \frac{\mu_1}{\tan(\theta + \phi)} \frac{t}{T} \frac{\rho}{r},$$
substituting from equation B.

If we put $K = \dfrac{\tan \phi}{\tan(\theta + \phi)}$, and $A = \dfrac{\mu_1}{\mu} \dfrac{t}{T} \dfrac{\rho}{r}$, in the last equation, and invert both members; then, since $\mu = \tan \phi$, there results:

$$\frac{M_s}{M} = \frac{1}{1 + KA} \qquad (3)$$

Hence general formula 3 is established, in which K and A are the same as indicated in formulas 4 and 5.

MAXIMUM COMBINED EFFICIENCY OF SCREW AND THRUST-BEARING

To take into account the effect of a thrust-bearing it is observed that the net efficiency is the product of formulas 2 and 3.

Thus, net efficiency =

$$\eta = \frac{M_t}{M} = \frac{\tan \theta}{\tan(\theta + \phi)} \cdot \frac{1}{1 + A \dfrac{\tan \phi}{\tan(\theta + \phi)}}.$$

Putting $n = \tan \theta$, and $\mu = \tan \phi$,

$$\eta = \frac{n - \mu n^2}{B + Cn}$$

if $B = \mu(1 + A)$ and $C = 1 - A\mu^2$.

Whence $$\frac{d\eta}{dn} = -\frac{\mu C n^2 + 2\mu B n - B}{(B + Cn)^2}.$$

The condition for a maximum or minimum efficiency is that the numerator vanish, infinity not applying. The critical value of n will be found to correspond to the required maximum or minimum value of η. The critical value is

$$n = -D \pm \sqrt{D^2 + (D \div \mu)} \text{ (use upper sign)}$$

where

$$D = \frac{\mu (1 + A)}{1 - A\mu^2}.$$

It should be carefully remarked that D and μ have like signs for all probable values of μ, so that when μ is negative, (—D) is positive, but the entire quantity under the radical is positive. Moreover, the upper sign should always be employed, as n cannot be negative.

To illustrate the uses of the formula for n, the critical values of n in Tables V–VI may be found, employing the data on pages 393 and 394.

For Table V we have, $\mu = 0.1$ and $A = 0.4$. Thus,

$$D = \frac{0.1\ (1.4)}{1 - 0.004} = \frac{0.14}{0.996} = +0.1406$$

$$D^2 = 0.0198$$

$$\frac{D}{\mu} = 1.406$$

$$D^2 + \frac{D}{\mu} = 1.426$$

$$\sqrt{D^2 + \frac{D}{\mu}} = 1.194$$

$$-D = -0.141$$

$$n = 1.053. \text{ (See Table V.)}$$

For Table VI we have, $\mu = -0.1$ and $A = 1.2$. Thus,

$$D = \frac{-0.1\ (2.2)}{1 - 0.012} = \frac{-0.22}{0.988} = -0.2227$$

$$D^2 = +0.0496$$

$$\frac{D}{\mu} = +2.227$$

$$D^2 + \frac{D}{\mu} = +2.277$$

$$\sqrt{D^2 + \frac{D}{\mu}} = +1.509$$

$$-D = +0.223$$

$$n = +1.732. \text{ (See Table VI.)}$$

The corresponding maximum net efficiencies for Tables V–VI can now be computed for the critical values of n in the usual manner, or by substituting in

$$\eta = \frac{n - \mu n^2}{B + Cn}$$

or
$$\frac{M_t}{M} = \frac{n - \mu n^2}{\mu(1 + A) + (1 - A\mu^2)n}$$

remembering that the reciprocal is taken for the case of Table VI.

In the discussion of a paper by James McBride,* the maximum combined efficiency of screw-threads and collar bearing is considered by Mr. Wilfred Lewis, the formula deduced for the best slope being

$$\cot \alpha = \phi + \sqrt{\frac{1 + \phi^2}{1 + n}}$$

where n = ratio of mean diameter of step, or collar, to mean diameter of thread, being the same as the ratio $\rho \div r$ herein.

ϕ = coefficient of friction, assumed the same for collar and threads, the condition corresponding to $\mu = \mu_1$, in present notation.

α = angle of slope, so that cot α is the reciprocal of n as employed herein.

It is further tacitly assumed that t = T, so that, in the assumptions of Mr. Lewis, n is the same as A in the present paper, when t = T and $\mu = \mu_1$.

Therefore, to show that the present formula for the best slope of threads for combined thread and thrust-bearing frictions represents the more restricted assumptions of Mr. Lewis, it is merely necessary to take the reciprocal of the formula for n above and to substitute n for A; ϕ for μ; 1 for $t \div T$; and n for $\rho \div r$.

Thus,
$$\cot \alpha = \frac{1}{n} = \frac{1}{\sqrt{D^2 + D \div \mu} - D}$$
$$= \frac{\sqrt{D^2 + D \div \mu} + D}{D \div \mu}$$
$$= \mu + \sqrt{\mu^2 + \mu \div D}$$

*"Some Experiments with a Screw Bolt." Transactions of the American Society of Mechanical Engineers, 1890-1891, v. 12, p. 781.

$$= \mu + \sqrt{\frac{1 + \mu^2}{1 + A}}$$

since $$\frac{\mu}{D} = \frac{1 - A\mu^2}{1 + A}$$

Therefore, $$\cot \alpha = \phi + \sqrt{\frac{1 + \phi^2}{1 + n}}$$

which shows the relation when the coefficients of thread and bearing frictions are equal and when the thrust on the collar and the thrust due to thread reactions are also equal.

Perhaps it should be remarked that it would be correct to employ sec α, as defined herein, as the factor of friction for V-threads instead of the angle β, as on page 786 of the discussion by Mr. Lewis. (See Fig. 2, herein.)

ILLUSTRATING USES OF THE FORMULAS AND TABLES

Tables I and II give the values of $M_t \div M_s$, $M_s \div M_t$, and K directly as functions of n and μ. Table I is employed when μ, and consequently ϕ, are positive. Table II must be employed when they are negative. The latter is the case when the weight is being let down or the screw slackened. A is calculated by formula 5 and is positive, since μ_1 and μ always have like signs. With the values of K and A thus obtained, the ratio $M_s \div M$, and consequently the efficiency of the pivot or thrust-bearing, may be either calculated by formula 3 or taken directly from Table Ia or IIa. The efficiency of the screw, including both thread friction and pivot friction, is the product of the efficiencies of the thread and pivot.

If the torque be communicated to the screw by means of a lever, and it is desired to include the friction due to the side-thrust of the lever on a neck journal, then the efficiency of the screw must be multiplied by the efficiency of the neck journal to obtain the net efficiency of the machine. Similarly we must multiply by the efficiency of any device which communicates torque—as, for example, bevel-gears—in order to obtain the net efficiency of the combination.

The series of values of $M_t \div M_s$ given in the tables have been discussed for maxima and minima for each of the coefficients

of friction appearing. The corresponding values of n, which give the maxima and minima of the ratio for given coefficients, have also been calculated. The formulas employed are:

$$(M_t \div M_s) \text{ max. or min.} = (\sec\phi - \tan\phi)^2 \qquad (7)$$

$$\text{Slope for max. or min.} = n_m = (\sec\phi - \tan\phi) = \tan\left(\frac{\pi}{4} - \frac{\phi}{2}\right) \qquad (8)$$

Therefore, the square of the best slope of thread is the numerical value of maximum thread efficiency, or its reciprocal, according as the screw delivers or receives thrust.

The results of these calculations have been recorded in Tables III and IV. Table IV is for the case of lowering the weight or slackening the screw, and was calculated by taking $\tan\phi$ negative in formulas 7 and 8.

The writer possesses a "spiral-ratchet" screw-driver which he purchased more than twenty years ago. This tool consists of a long quadruple-threaded screw of large relative pitch passing into a cavity of the handle through a deep nut at the bottom of the handle. The lower end of the screw "spiral" operates the chuck by means of a ratchet. When the handle is thrust downward the ratchet transmits the twist of the screw "spiral" to the head of the screw to be driven. This presents a case of practically pure thread friction. For the sake of illustrating the use of the formulas and tables, the design of this tool will be discussed as it affects mechanical purchase and efficiency.

Apparently the only object considered in the design was to secure sufficient purchase with such a thrust as an ordinary person would be likely to exert upon the handle. It further seems that this has been accomplished by giving the threads a very steep slope.

The dimensions of the screw "spiral" are approximately: $r = 0.13$ inch, $p = 1.45$ inches, $n = 1.78$. Assume $\mu = 0.15$. With these values of μ and n, remembering that μ is negative, we calculate $M_t \div M_s$ by formula 2 and find it to be $+1.39$. This means that the weight upon the handle is 1.39 times the theoretical weight necessary to produce the actual torque upon the screw were there no friction. Or, if we take the reciprocal of

$M_t \div M_s$, as explained under the definitions, we shall find that the mechanical efficiency of the tool is $1 \div 1.39 = 0.72$. That is, 72 per cent. of the work expended is effective in driving the screw. The work due to the settling of the driven screw is negligible.

Let us now see whether we can improve the efficiency without sacrificing purchase. Referring to Table IV, with $\mu = 0.15$ the corresponding values of $(M_t + M_s)_m$ and n_m are, respectively, 0.740 and 1.16. That is, by reducing the slope of the threads to 1.16 instead of 1.78, the efficiency may be improved. In order now to prevent loss of purchase, the pitch must remain unaltered. This is accomplished, without loss of efficiency, by increasing the radius in the ratio of 1.78 to 1.16, or to 0.199 inch; say to 0.2 inch.

Let the two designs be compared by supposing a thrust of 50 pounds upon the handle. By formula 1, the torque delivered to the screw spiral in either case is $50 \times 1.45 \div 6.28 = 11.6$ inch-pounds. In the actual design, 72 per cent. of this, or 8.32 inch-pounds, is delivered to the screw to be driven. In the new design, 74 per cent., or 8.58 inch-pounds, will be delivered. Hence we have improved both the efficiency and the mechanical purchase by altering the dimensions of the screw "spiral."

Mr. Kingsbury has called the author's attention to the interesting effect upon the efficiency and design of the screw caused by a reduction in the value of the coefficient of friction. In the foregoing problem, for example, if $\mu = 0.10$, instead of 0.15, the efficiency of the tool becomes 82 per cent. and the radius 0.21 inches. (See Table II.) Thus, by decreasing the coefficient 33–1/3 per cent., the efficiency is increased by only 11 per cent. of its former value. In this case Table IV gives $n = 1.105$.

Mr. Kingsbury says that 0.10 is probably a better value for the coefficient in the foregoing problem than 0.15, as the slipping speed is rather high, on the average. In this the author concurs as a general proposition, but states that the character of the surfaces in the particular tool involved was not such as would warrant the use of low values.

A screw having a pivot or thrust-bearing will now be examined for the purpose of further illustration. Side-thrust, however, will be omitted for the present.

The nut is rigidly attached to the weight to be lifted and runs up and down the vertical screw which rests upon a step or collar thrust-bearing. The weight to be lifted, together with the weight of the nut, is 100 pounds. The screw weighs 25 pounds. Hence the down thrust on the step is 125 pounds. That is, $t \div T = 1.25$. Suppose that $r = 1$ inch and $\rho = 0.4$ inch. Then $\rho \div r = 0.4$. The coefficient of friction for the threads may be 0.1 and the coefficient for the thrust-bearing slightly less, say 0.08. Hence $\mu_1 \div \mu = 0.8$. Suppose further that $n = 0.04$.

Making use of the tables the solution would be as follows: Calculate A by formula 5. Thus $A = 0.8 \times 0.4 \times 1.25 = 0.4$. Enter Table I in the column headed 0.04 on the line marked 0.1. The value 0.284 so found is the ratio $M_t \div M_s$ for the slope 0.04 and coefficient 0.1. Enter Table I again, but in the column headed 0.1 on the line marked 0.04. The value 0.710 so found is the value of K for the slope 0.04 and coefficient 0.1. Now enter Table Ia in the column headed 0.4 and by interpolation between the lines marked 0.7 and 0.8 determine 0.779, which is the value of the ratio $M_s \div M$ for the values 0.4 of A and 0.710 of K. The product of the two numbers thus taken from the tables, $0.284 \times 0.779 = 0.221$, is the efficiency of the screw as explained in the definitions.

To show the effect of changing the pitch of this screw, Table V has been calculated in the foregoing manner for various pitches corresponding to the slopes appearing in the first column.

All dimensions, other than the pitch, remain constant throughout. The point where the net efficiency reaches a maximum is indicated and the values for that point are included.

It must be remarked that the point of maximum net efficiency, including bearing friction due to thrust, does not coincide with the point of maximum thread efficiency. This may be verified by reference to Table III.

Of course the formulas may be employed instead of the tables, and when the conditions of a problem extend beyond the limits of the tables the formulas are indispensable.

Table VI has been calculated in a manner similar to that employed for Table V but by referring to Tables II and IIa instead of I and Ia. The dimensions and conditions were taken

as follows: $\mu = 0.1$, $\mu_1 = 0.08$, $t \div T = 1$, $\rho \div r = 1.5$, $A = 0.8 \times 1.5 = 1.2$. The values of the slope of the threads in the different cases appear in column 1 as before. In this case the nut is supposed to turn and let a weight settle.

The interpretation of this table is as follows: A negative net efficiency means that the slope of the threads is not sufficient to admit of the settling of the weight without the aid of an *assisting* torque upon the nut. So that the work of both the assisting torque and the settling weight is consumed in friction. A positive net efficiency, on the other hand, means that the slope of the thread is such that if the settling of the weight is not resisted by an *opposing* torque the resulting motion will be accelerated. At the point where the net efficiency is zero, the weight is in a state of unstable equilibrium, without any torque passing to or from the nut. In the case of a settling weight it may be shown that when the constituent efficiencies are respectively ∞ and 0, which is the case when $n = \mu$, the net efficiency evaluates to an equality with A. This condition has been indicated in the table.

If we now wish to ascertain the combined efficiency when the torque is communicated to the screw by means of some special device we may do so by multiplying the product of the previously obtained efficiencies by the efficiency of the device.

Let it be required to determine the efficiency of a hand-lever of length L which transmits the torque to the screw or nut through the medium of a neck journal whose average radius of bearing surface is R. Then the side-thrust upon the neck journal will depend upon the moment, M_L, of the torque applied to the hand-lever and the length of the lever. This side-thrust will be obtained by taking the quotient $M_L \div L$. The journal friction will be obtained by multiplying this side-thrust by the coefficient of journal friction, μ_2, and the moment of this friction will result by multiplying the last product by the radius, R, of the journal. Hence the moment of the journal friction will be $\mu_2 \frac{R}{L} M_L$.

Now this frictional torque will oppose or assist the torque M_L according as the motion of the lever is with or against the twist which M_L tends to create. Therefore the relation between the torque M which exists in the journal, the torque upon the hand-lever and the frictional torque must be

$M = M_L \mp \mu_2 \frac{R}{L} M_L$, where we are to take the minus sign when M_L produces the motion, but must take the plus sign when M is sufficient to overcome M_L.

Dividing the last equation through by M_L it becomes evident that the efficiency of the neck journal is

$$\text{either} \qquad \frac{M}{M_L} = 1 - \mu_2 \frac{R}{L} \qquad (9)$$

$$\text{or} \qquad \frac{M_L}{M} = \frac{1}{1 \mp \mu_2 \frac{R}{L}} \qquad (10)$$

Equation 9 applies particularly to the case of lifting a weight or overcoming a load by means of a screw. Equation 10 applies to the case of letting a weight down by means of a screw. Further, it is clear from what has been said that when the slope of the thread is not sufficient to admit of the settling of the weight without assistance—that is, when the screw "locks"—the minus sign must be employed, while if the screw does not lock, the plus sign must be employed.

We may now apply these formulas to the screws whose efficiencies are given in Tables V–VI. Evidently formula 9 applies without remark directly to Table V. Formula 10, however, needs more attention. It may be applied to Table VI if we make sure to take the correct sign. It will be remembered that the net efficiency in Table VI is negative when it is necessary to communicate an assisting torque, and that it is positive when the weight settles without assistance. It is, therefore, necessary to take the plus or minus sign according as the efficiency is plus or minus.

Tables Va–VIa give the resulting net efficiencies for the screws of Tables V–VI, respectively, when the efficiency of the hand-lever is applied. They will be understood without further remark.

The screws of Tables V, VI, Va, and VIa, however, are intended to be equivalent, as respects efficiency, to the screws given on page 30 of Dahlstrom's translation of "The Mechanics of Hoisting Machinery," by Weisbach and Herrmann, published by The Macmillan Company, 1907. On referring to this book it

will be seen that all the results of Table Va agree with those given by Weisbach, but that there is a discrepancy of about 4 per cent. in the results for Table VIa. If the writer correctly understands the conditions supposed by Weisbach, concerning the hand-lever, the sign of the denominator of the first fraction employed by him in the expression for the efficiency of the screw or nut when the weight is being lowered is incorrect.

We may now revert to the problem criticized in the opening paragraphs. Since it appears therein that the screw turns and bears upon a collar under the hand-wheel we may take $\rho \div r = 1.6$. The collar supports both the weight to be lifted and the weight of the screw, while the thrust upon the threads is 5000 pounds. Hence $t \div T = 5111 \div 5000 = 1.023$. Suppose $\mu = \mu_1 = 0.1$, and that $r = 1$, all the other conditions being as given. Then $A = 1.6 \times 1.023 = 1.64$; $n = 1 \div 6.28 = 0.159$; $M_t \div M_s = 0.605$; $K = 0.380$; $M_s \div M = 0.617$. Therefore, $M_t \div M = 0.373$, which is the efficiency. From formula 1 we have:

$$M_t = \frac{5000 \times 1}{6.28 \times 1} = 796 \text{ inch-pounds.}$$

Hence, excluding friction, the requisite force to be applied to the circumference of the hand-wheel is $796 \div 24 = 33.2$ pounds. Since the efficiency is only 0.373, we shall have $M = 2.68\ M_t$ and, consequently, the actual force necessary will be $2.68 \times 33.2 = 89$ pounds.

If all the dimensions and conditions remain as above except that the diameter of the screw is changed, so that $r = 2$ inches instead of 1 inch, it may be easily shown that the force required to be applied to the circumference of the hand-wheel is increased to 143 pounds. This shows that a general friction factor will not suffice in the treatment of screw friction.

APPLICATION TO V-THREADS

All the formulas and tables given above for square threads apply directly to V-threads, and angular threads in general, if in place of μ we introduce $\mu \sec \alpha$, where α is the angle between the normal to the axis of the screw, through the point of application of the resultant of thread friction, and a plane which is tangent to the surface of the thread at the same point.

The determination of the exact point of application of the resultant of thread friction would necessitate recourse to the principles of elasticity. It would not add greatly to confidence in results, since there is always doubt as to the exact value of the coefficient of friction. Hence, for practical purposes, the resultant of thread friction may be considered as acting at a distance, from the axis of the screw, equal to the arithmetical mean of the inner and outer radii of the threads.

It will be observed that α is not half the angle of the thread, measured at its edge in an axial plane. The angle which the surface of the thread makes with a normal to the axis measured in an axial plane is constant, while α is different for every different distance from the axis. In fact, if β is half the angle of the thread at its edge, made by an axial section, the relation between α and β is:

$$\text{Sec}\ \alpha^* = \sec\beta\ \sqrt{1 - (\sin\theta\sin\beta)^2}.$$

Here the modification of the method for square threads to that for V-threads differs from the method proposed by Reuleaux in his handbook of machine design, "The Constructor," as translated and published by Henry Harrison Suplee, New York, 1907. The difference consists merely in the use by Reuleaux of the angle β in the same manner in which α has been employed above. Where the angle θ is small, the error occasioned would be inconsiderable.

On page 187 of Unwin's "Elements of Machine Design," 1906, a formula for V-threads is given, in the denominator of which, if the author is not mistaken, cos β erroneously appears in place of sec β. It would further seeem that the angle β has the same definition as given by Reuleaux, whereas the present writer contends that it should be replaced by α as defined above. The propriety, then, of dropping a term in the denominator, as is done a few lines below on the same page, is not apparent.

In view of these facts it will be proper to give a proof of the accuracy of the method proposed herein for the adaptation, to V-threads, of the formulas and tables for square threads.

*Mr. Kingsbury has deduced $\sqrt{\sin^2\theta + (\cos^2\theta \div \cos^2\beta)}$ for the factor by which μ is to be multipled. This factor reduces easily to the value above given for sec α.

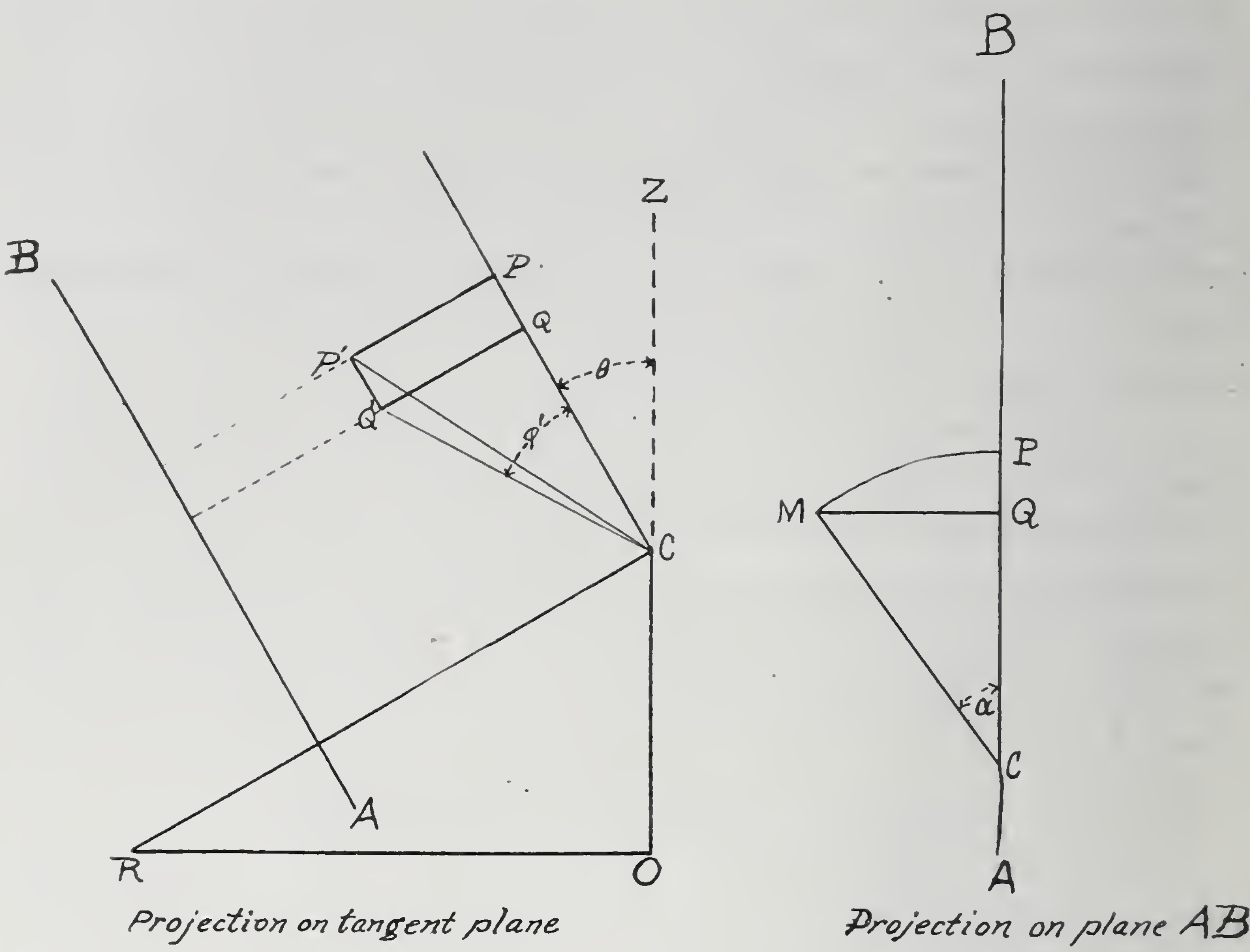


Fig. 2.

In Fig. 2 let the plane of the paper be parallel to the axis of the V-thread screw, and tangent to the axial cylinder passing through the point of application, C, of the resultant of thread friction on the surface of one of the threads. Then OZ, passing through C, is the projection of the axis. Let RC be the trace, on the paper, of the plane which is tangent to the thread at C.

If the threads were square, the last-mentioned plane would be perpendicular to the paper and all the forces acting upon a portion of the thread at C would lie in the plane of the paper, as in the demonstration concerned with Fig. 1.

In an axial section of the screw, however, the surfaces of the V-threads make an angle, β, with normals to the axis. This is one-half the angle of any thread at its edge in an axial plane. The plane which is tangent to the surface of the thread at the point C, therefore, makes some angle, α, with the plane perpendicular to the paper along its trace RC.

Consequently, the forces which act on the element C do not all lie in the plane of the paper. But, so far as equilibrium is concerned, the components of these forces which are perpendic-

ular to the paper are inoperative as regards torque or thrust. Therefore, the projection on the paper of the concurring forces at C will furnish the conditions as to torque and thrust.

Now the resultant of the friction and normal pressure which occur at C will always make the constant angle ϕ, for the same kind of surface, with the normal to the surface of the thread at C. If CM is that normal, then the angle ϕ is measured in the plane of CM and RC, since the friction is along RC.

Therefore, to find the angle ϕ', which the projection of the resultant of the pressure and friction at C makes with the projection of the normal to the surface of the thread at C, we have simply to find the projection of the angle ϕ upon the paper.

The angle ϕ lies in the plane of CM and CR. If we rotate CM to CP, we should have angle $PCP' = \phi$. The projection of ϕ on the paper would, therefore, be $QCQ' = \phi'$, in which $Q'Q = P'P$.

Whence $$Q'Q = CP \tan \phi$$

or $$\frac{Q'Q}{CQ} = \frac{CP}{CQ} \tan \phi.$$

But $$\frac{Q'Q}{CQ} = \tan \phi'$$

and $$\frac{CP}{CQ} = \sec \alpha$$

Therefore $$\tan \phi' = \sec \alpha \tan \phi.$$

It follows that the angle ϕ' may be employed in the case of V-threads in the same way that ϕ has been in the proof of the efficiency formula for square threads. This implies that we may substitute $\sec \alpha \tan \phi$ in place of $\tan \phi$, or, what is the same thing, $\mu \sec \alpha$ in place of μ in the formula for square threads, and obtain thereby the formula for V-threads; being careful to distinguish α and β.

Table VII gives the values of $\sec \alpha$ for different values of n and β.

WORM-GEARING

In the *American Machinist* for 1898, vol. 21, pages 24–26, 46–49, Halsey arrives at correct conclusions concerning the design of worm-gearing and gives rules which, if followed, should pro-

duce worms of high efficiency and long life. He does not provide means, however, for calculating the efficiency in terms of arbitrary dimensions. While it will hardly be within the scope of the present discussion to enter upon an analysis of worm-gearing, yet, with certain modifications, the formulas may be applied to such gearing, and a few remarks along that line may be of interest.

In the first place, it must be distinctly stated that a discussion of pure thread friction will not lead to the best design of a worm-carrying thrust. This point has been noticed by Halsey and is illustrated in the results of Table V, where it appeared that the maximum efficiency of a screw did not occur coincidently with the maximum efficiency of the threads. There is another loss of the same nature due to the side-thrust of the journal of the worm against its bearings by reason of the inclination and friction of the threads. No analysis of the worm will suffice, therefore, unless these losses due to thrust are included. The same remark does not apply to such a device as a hand-lever unless the coefficients of neck-journal friction vary with the power transmitted, or with the reaction at the threads.

There is another fact which seems to have been hitherto overlooked. It is that both velocity and friction are vectors—that is, directed magnitudes. Unless they act in the same line it is only the projection of one upon the other which is effective. Everyone is familiar with the way in which a prony brake floats upon the pulley. This is because the direction of the friction is nearly tangential to the pulley and, consequently, a very minute side-thrust will displace the brake. Another illustration of this is in the oil-pressure, gage-testing machine, where a weight is placed upon the top of a vertical plunger to transmit the pressure to the oil. A rotary motion of the plunger practically eliminates the vertical friction which, otherwise, would render the machine of little account. The cause finds itself in the fact that the motions produced by the side-thrust upon the prony brake and by the weight upon the plunger of the testing machine are very small relatively to the tangential velocities of the pulley and rotating plunger.

It follows from this that the combined thread friction and

tooth friction of the worm and wheel are to be added geometrically and not arithmetically. In other words, the combined friction is less than the arithmetical sum of the thread and tooth frictions.

With the foregoing statements in mind, the following observations may be made:

The efficiency of the threads depends only on the coefficient of friction, μ, and the slope of the threads. With a given slope, the pitch and radius of the worm do not influence this efficiency. It is only the ratio of the radius to the pitch which is operative. It follows that, with a given slope, no alteration of dimensions will affect the efficiency of the threads. Hence, one of the necessary conditions for a maximum efficiency of the worm is that A in equations 3 and 5 be taken as small as possible.

Since $A = \frac{\mu_1}{\mu}\frac{t}{T}\frac{\rho}{r}$ it can be made small only by such a design as will make one or more of the three ratios small: $\frac{\mu_1}{\mu}$ can be made small by means of a ball thrust-bearing; $\frac{t}{T}$ can be reduced by any device which reduces the thrust on the bearing as compared with the thrust delivered by the threads. A double right-and-left worm acting on two wheels would reduce A to zero. $\frac{\rho}{r}$ can be reduced by *increasing* the radius of the worm, by decreasing the radius of the thrust-bearing, or by both.

Having decided upon the best practical values for t, T, ρ, and r, the best slope for the threads may be determined by a process similar to that illustrated in Table V. The worm should then be cut to that slope, multiplying the number of threads if necessary.

It will be seen that this method gives slopes much larger than usually employed for the threads of worms, but where maximum efficiency is required this must be done as far as practicable. The best slope will be found much larger for low than for high coefficients of friction, as shown by Table III.

One other modifying influence must be taken into account. It is the side-thrust between the worm and the worm-wheel. This should be included in discussions for best design, as it operates in

very much the same way as the thrust on the collar or step. It is different in its effects from the hand-lever, etc.

The side-thrust of the wheel may be eliminated by employing right- and left-hand wheels and two worms. At the same time, with still two more wheels and worms, the friction losses in the step bearings may be eliminated by properly disposing the right- and left-hand worms and wheels in neutralizing pairs. This requires two worm-shafts, two wheel shafts and two counter-acting worms or wheels on each shaft. Incidentally, the journal friction due to the side-thrust of the worm journals might be reduced to the friction of rollers bearing upon each other, leaving practically pure thread friction.

By introducing two more worm-shafts, with two worms on each shaft engaging the sides of the wheels opposite the first two worm-shafts, the action may be made still more nearly perfect. Rolling disk wheels on the worm-shafts bear upon each other to eliminate side-thrusts of the worms. This arrangement consists of two wheel shafts, four wheels, four worm-shafts and eight worms, the efficiency being that of threads only. This arrangement eliminates side-thrusts upon the journal bearings of the wheels. Thus it is not necessary to introduce ball-bearings to secure pure thread friction, the efficiency of which need not be reduced by multiplication of the number of worms and wheels. The friction of the rolling disks would be negligible, with proper adjustment, and the device would be serviceable in transmitting power with high efficiency and might be employed in tests to determine the thread friction of worm-gears.

A more detailed discussion would require entirely too much space and is, therefore, reserved for the future.

Professor C. M. Allen and F. W. Roys have published the results of very interesting experiments upon the efficiency of gear drives, an abstract of their paper appearing in the *Journal* of The American Society of Mechanical Engineers, vol. 40, May 1918, page 367. The methods of testing are unique, and their results might be examined for values of the coefficients of friction for worm-gearing.

At the present moment the author does not have sufficient data concerning the ball-bearings of the worm-gear to give a

highly intelligent analysis, but submits the following:

The engineers of the test state that the gear was made of phosphor bronze with 40 teeth; pitch diameter, 10.5704 inches; throat diameter, 10.9964 inches; circular pitch, 0.8302 inch; angle of teeth with axis, 38° 16′ 5″; thickness of tooth, 0.3568 inch. The worm was made of Aurora steel, case-hardened, having nine teeth; pitch diameter, 3.015 inches; outside diameter, 3.441 inches; circular pitch, 1.0524 inches; angle of teeth with axis, 51° 43′ 55″; thickness of tooth, 0.295 inch; lead, 7.4719 inches. The drive was made by the Brown & Sharpe Manufacturing Company, who mounted it in a ball-bearing case especially designed for the purpose of testing. The results of the test were:

DATA OF EFFICIENCY TEST OF WORM-GEAR DRIVE

TEXAS COMPANY'S THUBAN OIL LUBRICATION

R.p.m. of motor	Scale pan, lb.	P_3 lb.	0.1837 P_3	× ft.	1.647 ×	P_1	Input, hp_{1000}
1109	5	200	36.73	1.72	2.83	39.56	19.78
1109	5	200	36.73	1.88	3.09	39.82	19.91
1137	4	160	29.38	1.42	2.34	31.72	15.86
1137	4	160	29.38	1.48	2.44	31.82	15.91
1156	3	120	22.03	1.23	2.03	24.06	12.03
1156	3	120	22.03	1.10	1.81	23.84	11.92
1175	2	80	14.69	0.92	1.515	16.205	8.102
1175	2	80	14.69	0.82	1.35	16.04	8.02
1193	1	40	7.345	0.70	1.15	8.46	4.23
1193	1	40	7.345	0.53	0.872	8.217	4.11
1200	½	20	3.673	0.68	1.12	4.793	2.39
1200	½	20	3.673	0.45	0.740	4.413	2.20

R.p.m. of motor	Input, actual hp.	Rider hp_{1000}	Hp. loss	Output, hp.	Efficiency, per cent.	Temp., degrees F.
1109	21.96	1.65	1.93	20.03	91.3	80
1109	22.10	1.90	2.11	19.99	90.5	150
1137	18.03	1.44	1.635	16.395	90.7	80
1137	18.10	1.50	1.706	16.394	90.6	150
1156	13.91	1.24	1.435	12.475	89.7	80
1156	13.75	1.10	1.270	12.480	90.8	150
1175	9.52	0.94	1.105	8.413	88.3	80
1175	9.42	0.82	0.964	8.456	89.9	150
1193	5.05	0.70	0.835	4.215	83.5	80
1193	4.90	0.54	0.645	4.255	86.8	150
1200	2.76	0.68	0.807	1.953	70.7	80
1200	2.65	0.46	0.554	2.096	79.1	150

$$\text{Therefore, thread slope} = \tan\theta = \frac{\text{lead}}{\text{circ. of pitch circle}}$$

$$= \frac{7.47}{9.47} = n = 0.789.$$

Assuming frictionless bearings, the efficiency from certain of the above tests gives the values of the coefficient of friction by Table I, or Diagram 1, as follows:

Input hp.	R. p. m.	Efficiency per cent.	Virtual coefficient μ
20	1109	90.9	0.0467
5	1193	85.1	0.0788
2.7	1200	79.9	0.1123

Average temperature = ½ (80 + 150) = 115 degrees F. Of course, the above results might be different for a different lubricant.

By reference to Table III it will be seen that, to secure maximum efficiency for any of the given conditions of test, the slope of thread should be changed as indicated below:

Input hp.	Virtual coefficient	Best slope of thread	Maximum efficiency per cent.
20	0.0467	0.9543	91.13
5	0.07878	0.9241	85.42
2.7	0.1123	0.8939	79.91

Apparently there is little to be gained by increasing thread slope, but it is clear that a tendency toward positive improvement can be made by such an alteration of design when the given lubricant is retained, and that the alteration may be greater the greater the load, at least up to 20 horsepower. If there is any collateral advantage to be gained by increased slope, such as increased speed of worm-wheel, then it would appear that an increase in thread slope is positively desirable.

If the efficiency of the ball-bearings were known it seems probable that the slope might be increased still more, since coefficients of thread friction would very likely be reduced. This is shown by the fact that the points of maxima move toward the right on Diagram 1 as the coefficient of friction diminishes.

APPENDIX

KINGSBURY'S COEFFICIENTS*

The tests were made upon a set of square-threaded screws and nuts of the following dimensions:

Outside diameter of screw........	1.426	inch
Inside diameter of nut............	1.278	"
"Mean diameter" of thread........	1.352	"
Pitch of thread..................	1/3	"
Depth of nut......................	1–1/16	" (effective)

This depth of nut makes the area of thread approximately one square inch, so that the total axial load on the screw is also the pressure per square inch on the thread surface.

The nuts fit the screws very loosely, so that all friction is excluded, except that on the faces of the threads directly supporting the load. The threads were cut carefully in the lathe, and had been worn to good condition by trials previous to those here recorded. Screw No. 5 was not quite so smooth as the others.

. . .

The screws and nuts used in the test were as follows:

SCREWS

No.	Material
1..................	Mild Steel.
2..................	Common Wrought Iron.
3..................	Cast Iron.
4..................	Cast Bronze.
5..................	Mild Steel—Case-hardened.

NUTS

6..................	Mild Steel.
7..................	Common Wrought Iron.
8..................	Cast Iron.
9..................	Cast Brass.

*From "Experiments on the Friction of Screws," by Albert Kingsbury. Transactions of the American Society of Mechanical Engineers, 1896, v. 17, pp. 96, 99, 103–105.

Four sets of tests were made with lubricants and pressures, as follows:

No. of set	Lubricant	Maximum load
1	Heavy Machinery Oil	14 000 lbs.
2	Winter Lard Oil.	14 000 "
3	Heavy Machinery Oil and Graphite, in equal volumes.	14 000 "
4	Heavy Machinery Oil.	4 000 "

The "Heavy Machinery Oil" was a purely mineral oil of specific gravity .912. The "Winter Lard Oil" had a specific gravity of .919.

. . .

TABLE I

MEAN COEFFICIENTS FOR HEAVY MACHINERY OIL

(Actually read at 10 000 pounds pressure per square inch. Each figure is the average for eight cards.)

Screws		Nuts 6 Mild steel	7 Wrought iron	8 Cast iron	9 Cast brass
1	Mild steel	.141	.16	.136	.136
2	Wrought iron	.139	.14	.138	.147
3	Cast iron	.125	.139	.119	.171
4	Cast bronze	.124	.135	.172	.132
5	Mild steel, case-hardened	.133	.143	.13	.193

Mean of all, .1426.

Highest for a single card (screw 5, nut 9) 20

Lowest for a single card (screw 3, nut 8) 11

TABLE II

MEAN COEFFICIENTS FOR LARD OIL

(Actually read at 10 000 pounds pressure per square inch. Each figure is the average for four cards.)

Screws	Nuts 6	7	8	9
1	.12	.105	.10	.11
2	.1125	.1075	.10	.12
3	.10	.10	.095	.11
4	.1150	.10	.11	.1325
5	.1175	.0975	.105	.1375

Mean of all, .1098.

Highest for a single card (screw 4, nut 9) 25

Lowest for a single card (screw 3, nut 8) 09

TABLE III

MEAN COEFFICIENTS FOR HEAVY MACHINERY OIL AND GRAPHITE

(Actually read at 10 000 pounds pressure per square inch. Each figure is the average for four cards.)

No. of screws	Nuts 6	7	8	9
1	.111	.0675	.065	.04
2	.089	.07	.075	.055
3	.1075	.071	.105	.059
4	.071	.045	.044	.036
5	.1275	.055	.07	.035

Mean of all, .07.

Highest for a single card (screw 5, nut 6) ..... .15
Lowest for a single card (screw 5, nut 9) ..... .03

TABLE IV

MEAN COEFFICIENTS FOR HEAVY MACHINERY OIL

(Actually read at 3000 pounds pressure per square inch. Each figure is the average for four cards.)

No. of screws	No. of nuts 6	7	8	9
1	.147	.156	.132	.127
2	.15	.16	.15	.117
3	.15	.157	.14	.12
4	.127	.13	.13	.14
5	.155	.1775	.1675	.1325

Mean of all, .1433.

Highest for a single card (screw 5, nut 7) ..... .19
Lowest for a single card (screw 2, nut 9) ..... .11

The conclusions which the results seem to warrant are:

That for metallic screws in good condition, turning at extremely low speeds, under any pressure up to 14 000 pounds per square inch of bearing surface, and freely lubricated before application of the pressure, the following coefficients of friction may be used:

COEFFICIENTS OF FRICTION

Lubricant	Minimum	Maximum	Mean
Lard oil	.09	.25	.11
Heavy machinery oil (mineral)	.11	.19	.143
Heavy machinery oil and graphite, in equal volumes.	.03	.15	.07

The writer does not consider that the tests prove that any one of the metals used develops less friction than any of the others, under the methods of testing employed, although such results might be inferred from Table III, for instance, in which the coefficients for the brass nut are uniformly lower than for any of the others. Nor does he believe that the method of testing employed is the best possible; a number of cast-iron nuts and screws tested by themselves, and a number of steel nuts and screws similarly tested, might give results showing less variation than is evident in the records given above, and hence more definitely comparable with each other.

For handy calculations concerning standard machine bolts Mr. Kingsbury recommends the formula:

$$\frac{Pd}{5} = M \text{ inch-pounds}$$

where P is the load in pounds, d the outside diameter of the thread in inches, and M the moment in inch-pounds. This includes the friction at face of nut and at screw threads.

TABLES AND DIAGRAMS

The tables have been explained in the text. It is thought that the reader will have no difficulty in applying the diagrams, and it is suggested that they be applied first to the problems given on pages 387–396.

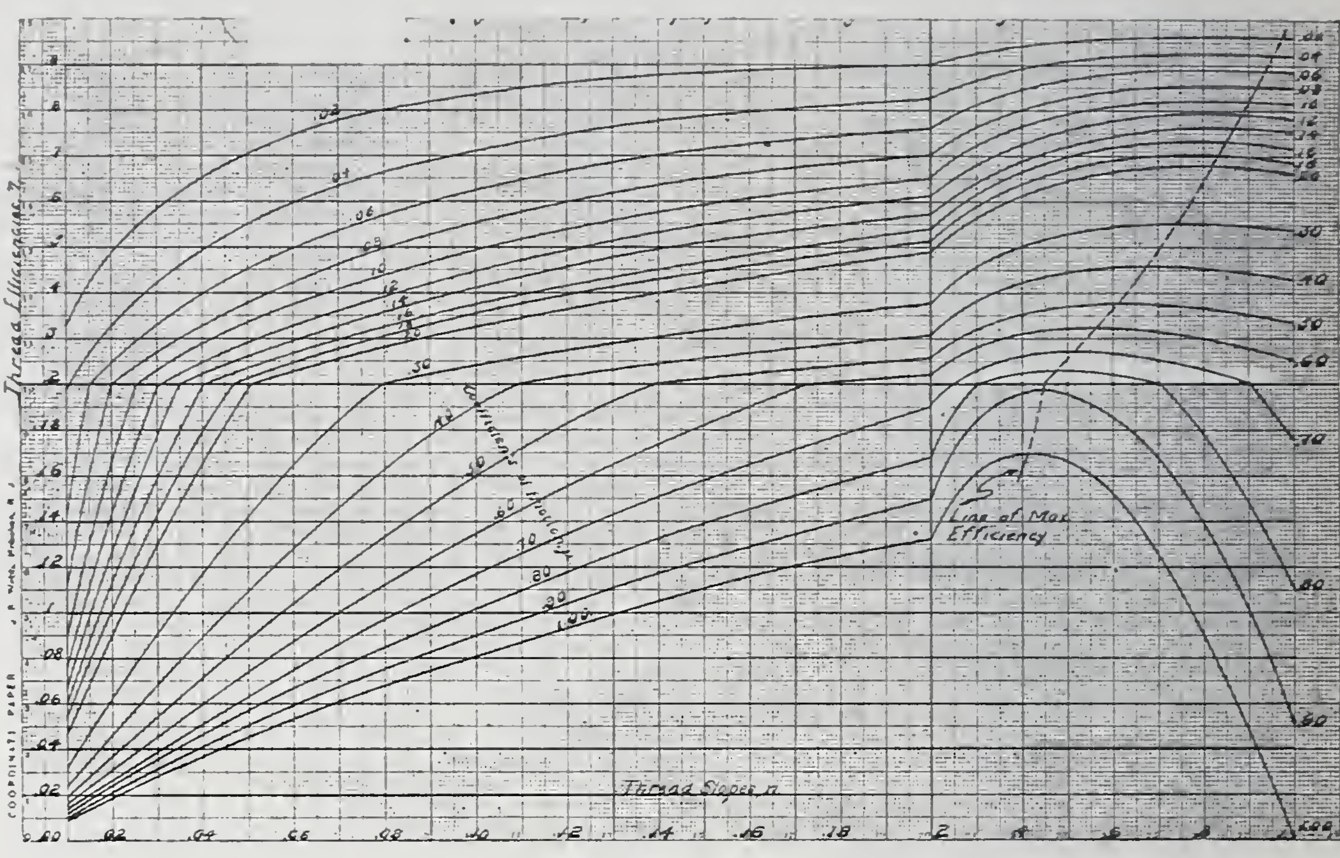
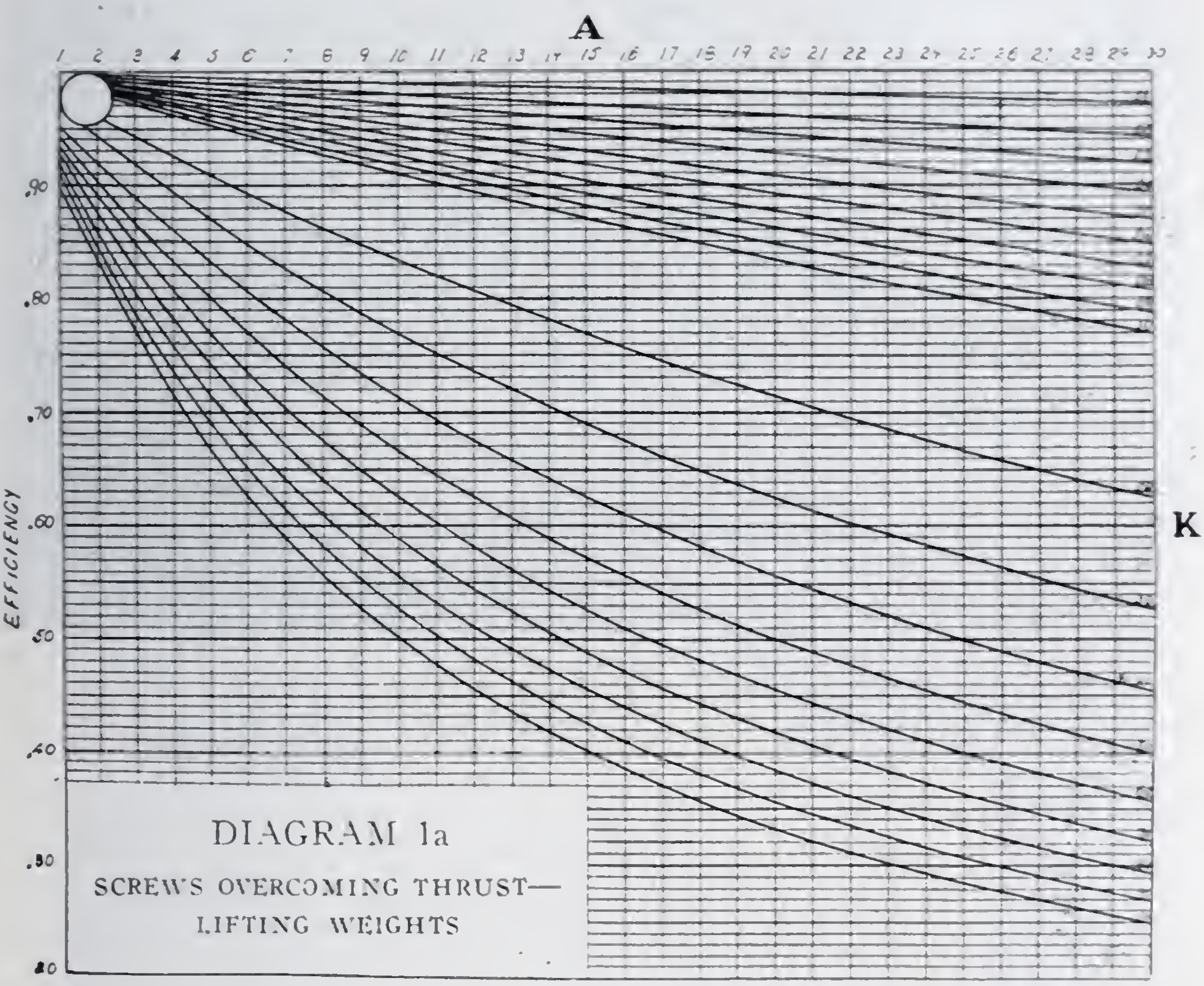


DIAGRAM 1

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

Values of the efficiency of square screw threads ($M_t \div M_s$) for various coefficients of friction, μ, and slopes of threads, n. By interchanging the values of μ and n, the diagram gives the values of K ($= 1 \div K^1$) with which to enter Diagram 1a, for the purpose of finding the efficiency of the bearing or step.



Values of the efficiency of thrust-bearing for screws ($M_s \div M$) for various values of K as determined by Diagram 1, and of A as determined by Formula 5.

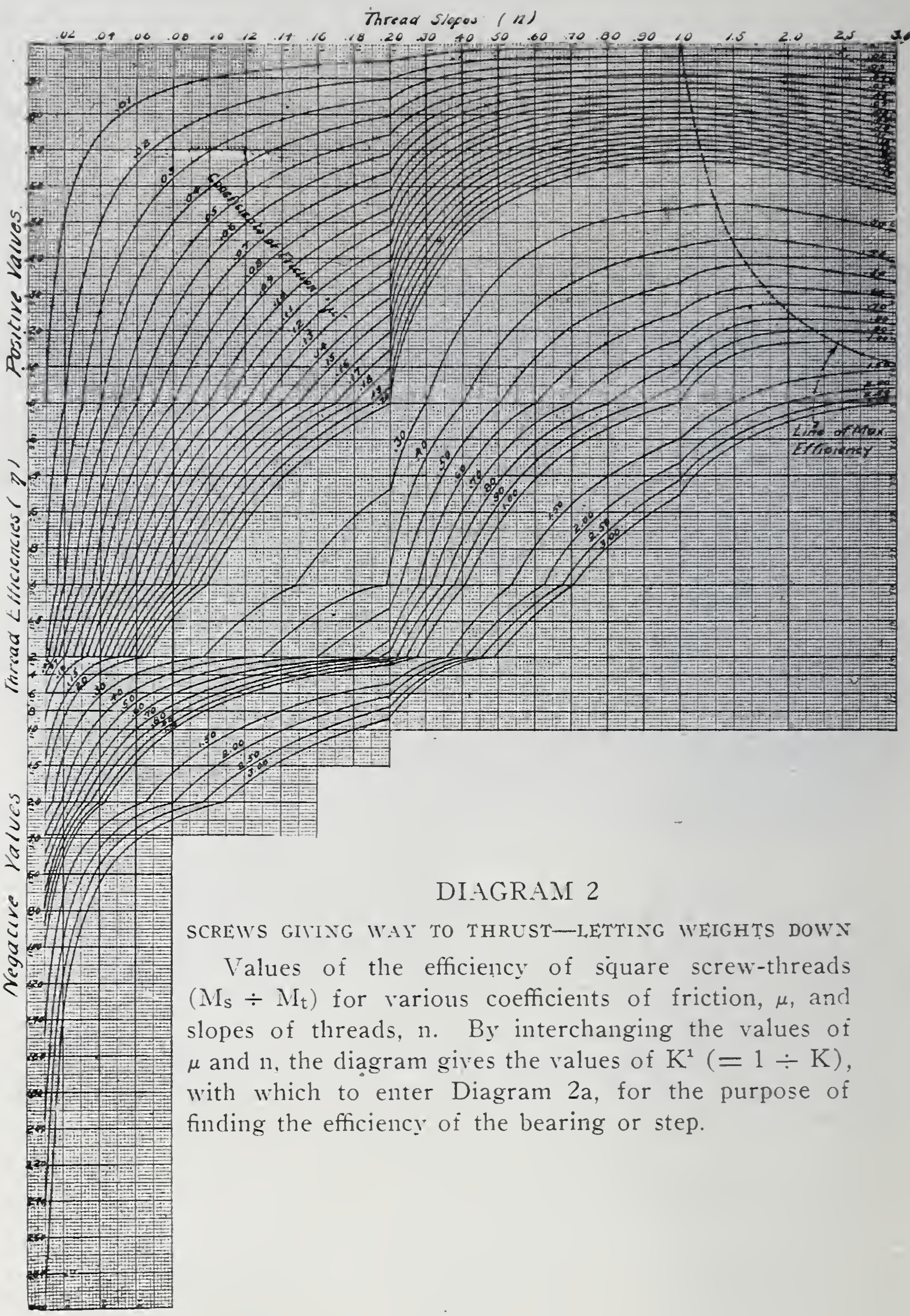


DIAGRAM 2

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

Values of the efficiency of square screw-threads ($M_s \div M_t$) for various coefficients of friction, μ, and slopes of threads, n. By interchanging the values of μ and n, the diagram gives the values of K^1 ($= 1 \div K$), with which to enter Diagram 2a, for the purpose of finding the efficiency of the bearing or step.

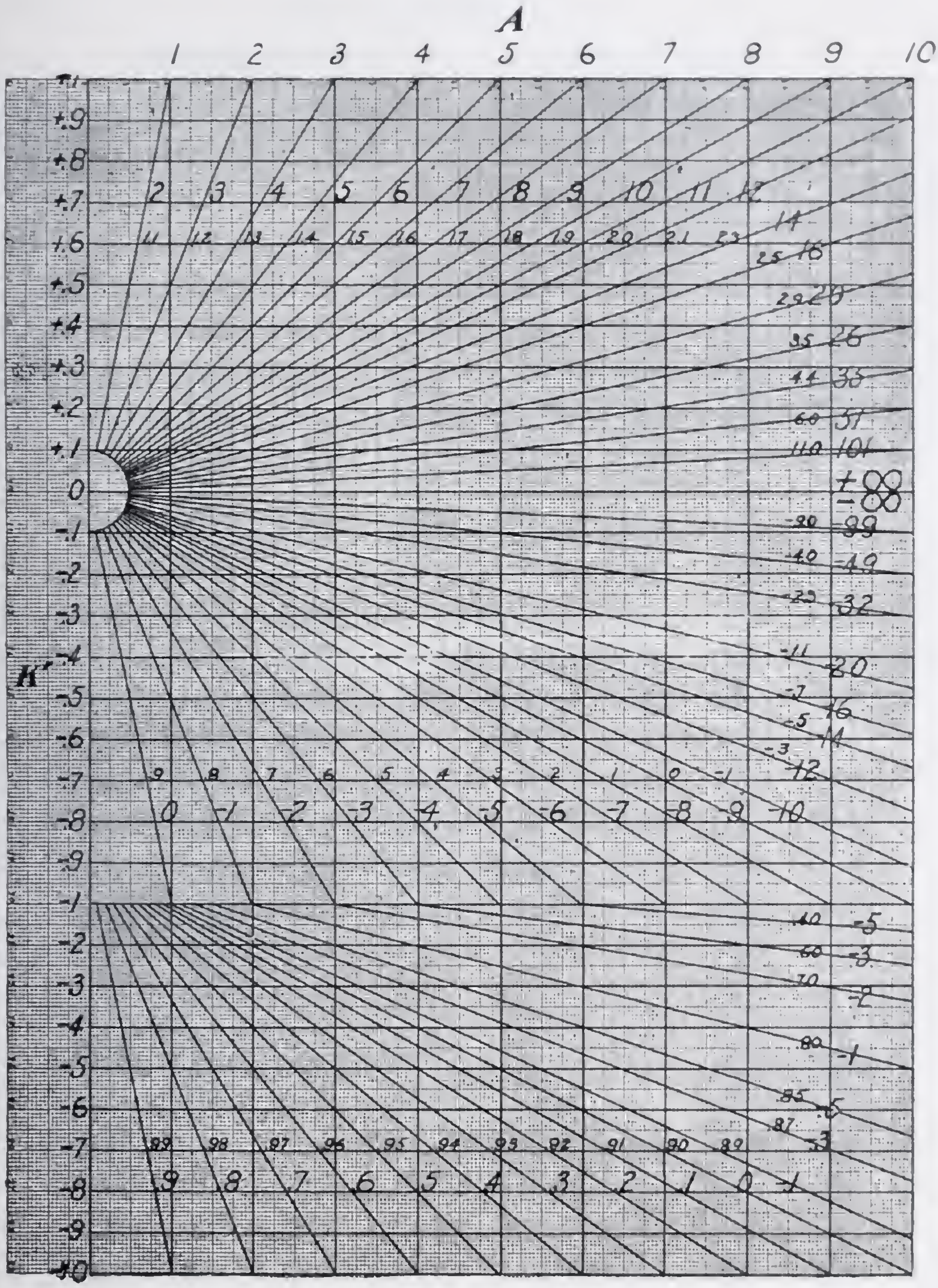


DIAGRAM 2a

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

Values of the efficiency of thrust-bearings for screws ($M \div M_s$) for various values of K^1 ($= 1 \div K$) as determined by Diagram 2, and of A as determined by Formula 5.

The scales on the diagonals in smaller figures are to be employed with either one-tenth the A scale or ten times the K^1 scale.

TABLE I

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

Values of the efficiency of square screw-threads ($M_t \div M_s$) for various coefficients of friction, μ, and slopes of threads, n. By interchanging the values of μ and n the table gives the value of K with which to enter Table Ia. (Calculated by slide rule.)

	n (for K interchange n and μ)								
μ	.01	.02	.03	.04	.05	.06	.07	.08	.09
.01	.490	.660	.747	.794	.829	.855	.873	.885	.900
.02	.332	.498	.598	.663	.712	.748	.777	.800	.818
.03	.249	.399	.500	.570	.624	.666	.700	.727	.750
.04	.198	.333	.427	.498	.554	.600	.636	.661	.687
.05	.166	.285	.374	.442	.500	.545	.583	.611	.639
.06	.142	.249	.333	.400	.455	.500	.538	.568	.596
.07	.125	.222	.300	.364	.417	.462	.500	.530	.559
.08	.111	.200	.273	.331	.382	.426	.464	.497	.523
.09	.100	.182	.250	.305	.355	.398	.435	.465	.494
.10	.091	.166	.231	.284	.331	.373	.410	.442	.469
.11	.083	.154	.214	.265	.310	.351	.387	.417	.446
.12	.077	.143	.199	.248	.292	.332	.365	.396	.425
.13	.071	.132	.188	.234	.276	.314	.347	.378	.404
.14	.067	.125	.176	.221	.262	.297	.332	.359	.386
.15	.062	.117	.166	.210	.248	.283	.316	.344	.370
.16	.058	.110	.157	.198	.236	.270	.300	.329	.354
.17	.055	.105	.149	.189	.225	.259	.288	.315	.341
.18	.0526	.0995	.1422	.1802	.2155	.248	.277	.303	.328
.19	.0497	.0948	.136	.172	.207	.237	.265	.292	.316
.20	.0474	.0905	.130	.165	.198	.228	.256	.281	.304
.30	.0321	.0621	.0901	.116	.141	.164	.185	.205	.224
.40	.0243	.0473	.0690	.0895	.109	.127	.145	.161	.177
.50	.0195	.0381	.0558	.0726	.0887	.104	.118	.132	.146
.60	.0163	.0318	.0468	.0610	.0746	.0876	.100	.112	.123
.70	.0140	.0274	.0402	.0525	.0643	.0756	.0864	.0968	.107
.80	.0122	.0240	.0353	.0461	.0562	.0664	.0760	.0850	.0938
.90	.0109	.0214	.0314	.0410	.0503	.0591	.0676	.0757	.0836
1.00	.0098	.0192	.0282	.0369	.0453	.0532	.0608	.0682	.0752

TABLE I—(Continued)

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

	n (for K interchange n and μ)									
μ	.10	.11	.12	.13	.14	.15	.16	.17	.18	.19
.01	.909	.916	.923	.929	.933	.937	.940	.944	.946	.947
.02	.833	.846	.857	.861	.875	.878	.884	.890	.895	.901
.03	.769	.785	.794	.807	.819	.828	.838	.846	.853	.860
.04	.710	.728	.745	.760	.773	.785	.792	.802	.811	.819
.05	.663	.683	.702	.718	.733	.742	.755	.766	.776	.787
.06	.621	.644	.663	.677	.694	.708	.721	.733	.744	.751
.07	.585	.608	.625	.644	.660	.676	.687	.700	.712	.720
.08	.550	.573	.594	.614	.638	.644	.658	.669	.682	.694
.09	.521	.544	.566	.583	.601	.617	.630	.644	.657	.667
.10	.495	.519	.538	.558	.576	.591	.606	.618	.632	.642
.11	.472	.493	.515	.533	.551	.568	.582	.597	.608	.619
.12	.449	.472	.492	.512	.530	.546	.559	.574	.584	.598
.13	.429	.451	.472	.491	.509	.525	.541	.554	.566	.580
.14	.412	.433	.454	.474	.492	.507	.521	.535	.549	.560
.15	.394	.417	.436	.454	.473	.488	.503	.518	.531	.543
.16	.379	.400	.420	.439	.456	.472	.486	.502	.514	.526
.17	.364	.386	.405	.427	.440	.457	.472	.486	.499	.511
.18	.351	.372	.391	.409	.427	.442	.457	.471	.484	.496
.19	.338	.358	.377	.396	.351	.429	.442	.457	.470	.482
.20	.327	.347	.366	.384	.400	.416	.430	.444	.457	.468
.30	.243	.260	.275	.290	.305	.318	.331	.344	.355	.365
.40	.192	.206	.220	.233	.245	.256	.268	.278	.288	.297
.50	.158	.170	.182	.193	.204	.213	.223	.232	.241	.249
.60	.134	.145	.155	.164	.173	.182	.190	.198	.206	.213
.70	.116	.125	.134	.142	.150	.158	.165	.172	.179	.185
.80	.102	.110	.118	.125	.132	.139	.145	.151	.157	.163
.90	.0910	.0980	.105	.112	.118	.124	.129	.134	.140	.144
1.00	.0825	.0882	.0942	.100	.106	.111	.116	.121	.125	.129

TABLE I—(Continued)

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

	n (for K interchange n and μ)								
μ	.20	.30	.40	.50	.60	.70	.80	.90	1.00
.01	.948	.964	.971	.975	.978	.979	.980	.981	.974
.02	.905	.932	.946	.952	.956	.959	.960	.961	.961
.03	.866	.901	.920	.930	.935	.939	.941	.941	.941
.04	.827	.872	.895	.907	.915	.919	.922	.923	.923
.05	.791	.845	.872	.887	.896	.901	.903	.905	.905
.06	.760	.818	.850	.867	.876	.882	.885	.887	.886
.07	.730	.793	.828	.846	.859	.864	.868	.869	.869
.08	.702	.770	.806	.828	.839	.846	.849	.852	.853
.09	.676	.748	.788	.810	.822	.830	.834	.836	.835
.10	.653	.728	.768	.791	.806	.813	.818	.819	.818
.11	.631	.707	.749	.774	.789	.797	.804	.803	.802
.12	.610	.688	.732	.758	.774	.781	.785	.787	.786
.13	.590	.670	.716	.742	.758	.776	.773	.771	.770
.14	.571	.652	.699	.727	.743	.752	.756	.757	.755
.15	.554	.637	.683	.711	.728	.737	.741	.741	.739
.16	.538	.630	.669	.696	.714	.723	.726	.727	.724
.17	.522	.606	.653	.683	.695	.709	.712	.712	.709
.18	.508	.572	.640	.669	.687	.696	.698	.698	.695
.19	.493	.577	.626	.655	.673	.682	.685	.684	.680
.20	.480	.564	.614	.643	.660	.669	.672	.670	.667
.30	.376	.455	.503	.531	.547	.553	.552	.548	.539
.40	.307	.377	.420	.444	.456	.458	.454	.443	.429
.50	.257	.319	.346	.375	.382	.379	.369	.354	.333
.60	.220	.273	.304	.318	.320	.312	.297	.276	.250
.70	.191	.237	.262	.271	.268	.255	.234	.208	.176
.80	.168	.207	.227	.231	.223	.205	.180	.148	.111
.90	.149	.182	.197	.196	.184	.174	.132	.0951	.0527
1.00	.133	.162	.171	.167	.150	.124	.0888	.0474	0

TABLE Ia

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

Values of the efficiency of thrust-bearings for screws ($M_s \div M$) for various values of K as determined by Table I, and of A as determined by Formula 5. (Calculated by slide rule.)

$$\text{Formula 5 is:}\quad A = \frac{\mu_1}{\mu}\,\frac{t}{T}\,\frac{\rho}{r}$$

	A									
K	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.01	.999	.998	.998	.997	.997	.995	.994	.993	.992	.991
.02	.998	.997	.995	.993	.991	.989	.987	.985	.983	.981
.03	.998	.995	.992	.989	.986	.983	.980	.978	.975	.972
.04	.997	.993	.989	.985	.981	.978	.974	.969	.965	.962
.05	.996	.991	.986	.981	.976	.972	.967	.962	.957	.953
.06	.995	.989	.983	.978	.972	.965	.960	.955	.949	.944
.07	.994	.987	.980	.974	.967	.960	.954	.946	.941	.935
.08	.993	.985	.978	.969	.962	.955	.946	.940	.933	.927
.09	.992	.983	.975	.965	.957	.949	.941	.933	.925	.918
.10	.991	.981	.972	.962	.953	.944	.935	.927	.918	.910
.20	.981	.962	.944	.927	.010	.893	.878	.863	.848	.834
.30	.972	.944	.918	.893	.870	.848	.827	.807	.788	.770
.40	.962	.927	.893	.863	.834	.807	.782	.758	.736	.715
.50	.953	.910	.870	.834	.800	.770	.741	.715	.690	.667
.60	.944	.893	.848	.807	.770	.736	.705	.676	.649	.625
.70	.935	.878	.827	.782	.741	.705	.672	.641	.614	.588
.80	.927	.863	.807	.758	.715	.676	.641	.610	.582	.556
.90	.918	.848	.788	.736	.690	.649	.614	.582	.553	.526
1.00	.910	.834	.770	.715	.667	.625	.588	.556	.526	.500

TABLE Ia—(Continued)

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

	A									
K	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0
.01	.990	.989	.988	.987	.986	.985	.984	.983	.982	.981
.02	.979	.978	.975	.974	.972	.969	.968	.965	.964	.962
.03	.968	.965	.963	.960	.957	.955	.952	.949	.946	.944
.04	.958	.955	.951	.946	.944	.940	.937	.933	.929	.927
.05	.948	.944	.939	.935	.931	.927	.922	.918	.913	.910
.06	.939	.933	.928	.923	.918	.912	.907	.903	.898	.893
.07	.929	.923	.916	.911	.905	.899	.894	.888	.883	.878
.08	.920	.912	.906	.899	.893	.887	.880	.874	.868	.863
.09	.911	.903	.895	.888	.881	.874	.867	.861	.854	.848
.10	.902	.893	.885	.878	.870	.863	.854	.848	.841	.834
.20	.820	.801	.794	.782	.770	.758	.746	.736	.725	.715
.30	.752	.736	.720	.705	.690	.676	.662	.649	.637	.625
.40	.694	.676	.656	.641	.625	.610	.595	.582	.568	.556
.50	.645	.625	.606	.588	.572	.556	.541	.526	.513	.500
.60	.603	.582	.562	.544	.526	.510	.495	.481	.468	.455
.70	.565	.544	.525	.505	.488	.472	.457	.443	.429	.417
.80	.532	.510	.490	.472	.455	.439	.424	.410	.397	.385
.90	.503	.481	.461	.443	.426	.410	.395	.382	.369	.357
1.00	.476	.455	.435	.417	.400	.385	.371	.357	.345	.333

TABLE Ia—(Continued)

SCREWS OVERCOMING THRUST—LIFTING WEIGHTS

					A					
K	2.1	2.2	2.3	2.4	2.5	2.6	2.7	2.8	2.9	3.0
.01	.980	.979	.978	.978	.977	.975	.975	.974	.973	.972
.02	.960	.958	.956	.955	.953	.951	.949	.946	.945	.944
.03	.941	.939	.936	.933	.931	.928	.925	.923	.920	.918
.04	.923	.920	.916	.912	.910	.906	.903	.899	.895	.893
.05	.905	.902	.898	.893	.889	.885	.881	.878	.874	.870
.06	.888	.884	.879	.874	.869	.864	.861	.856	.852	.848
.07	.872	.867	.861	.856	.851	.846	.841	.836	.831	.827
.08	.856	.850	.845	.839	.834	.828	.823	.818	.812	.807
.09	.841	.835	.829	.823	.816	.810	.805	.799	.793	.788
.10	.827	.820	.814	.807	.800	.794	.788	.781	.776	.770
.20	.705	.694	.685	.676	.667	.656	.649	.641	.633	.625
.30	.614	.603	.592	.582	.572	.562	.553	.544	.535	.526
.40	.544	.532	.521	.505	.500	.490	.481	.472	.463	.455
.50	.488	.476	.465	.455	.445	.435	.426	.417	.408	.400
.60	.443	.431	.421	.410	.400	.391	.382	.377	.365	.357
.70	.405	.394	.387	.377	.364	.355	.346	.338	.330	.323
.80	.377	.362	.354	.343	.338	.325	.317	.309	.301	.294
.90	.346	.336	.326	.317	.308	.299	.292	.284	.277	.270
1.00	.323	.313	.303	.294	.286	.278	.270	.263	.256	.250

TABLE II

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

Values of the efficiency of square screw-threads ($M_s \div M_t$) for various coefficients of friction, μ, and slopes of threads, n. By interchanging the values of μ and n the table gives the values of K^1 ($= 1 \div K$) with which to enter Table IIa. (Calculated by slide rule.) Negative values of K^1 below diagonal line of zeros.

	n (For K^1 interchange n and μ)									
μ	.01	.02	.03	.04	.05	.06	.07	.08	.09	.10
.01	0	.500	.666	.750	.799	.833	.857	.874	.888	.899
.02	.999	0	.333	.499	.599	.666	.712	.749	.777	.798
.03	1.99	.500	0	.250	.399	.499	.570	.624	.666	.698
.04	2.99	.999	.333	0	.199	.333	.428	.499	.554	.598
.05	3.99	1.50	.666	.250	0	.166	.285	.374	.443	.498
.06	4.99	2.00	.999	.499	.199	0	.142	.249	.332	.398
.07	5.99	2.50	1.33	.749	.399	.166	0	.124	.221	.298
.08	6.99	3.00	1.66	.998	.598	.332	.142	0	.110	.198
.09	7.99	3.49	1.99	1.24	.798	.497	.284	.124	0	.099
.10	8.99	3.99	2.32	1.49	.996	.663	.426	.248	.110	0
.11	9.99	4.49	2.65	1.74	1.19	.828	.567	.372	.220	.099
.12	10.98	4.99	2.99	1.99	1.39	.993	.709	.496	.330	.197
.13	11.97	5.49	3.32	2.24	1.59	1.16	.850	.619	.440	.296
.14	12.96	5.99	3.66	2.48	1.79	1.32	.992	.742	.549	.395
.15	13.96	6.49	3.99	2.73	1.98	1.49	1.13	.865	.658	.493
.16	14.96	6.98	4.32	2.98	2.18	1.65	1.27	.989	.767	.590
.17	15.96	7.48	4.64	3.23	2.38	1.82	1.42	1.11	.875	.688
.18	16.95	7.98	4.97	3.48	2.58	1.98	1.56	1.23	.985	.788
.19	17.95	8.48	5.31	3.73	2.78	2.15	1.70	1.35	1.09	.884
.20	18.95	8.98	5.63	3.97	2.98	2.30	1.83	1.48	1.20	.981
.30	28.90	13.91	8.93	6.45	4.93	3.93	3.21	2.69	2.27	1.94
.40	38.80	18.85	12.20	8.86	6.87	5.53	4.58	4.43	3.32	2.88
.50	48.70	23.75	15.43	11.3	8.79	7.12	5.94	5.05	4.36	3.81
.60	58.60	28.65	18.67	13.7	10.7	8.68	7.28	6.20	5.38	4.72
.70	68.50	33.55	21.87	16.05	12.56	10.24	8.58	7.34	6.38	5.61
.80	78.40	38.40	25.07	18.40	14.40	11.75	9.88	8.46	7.35	6.48
.90	88.20	43.20	28.23	20.75	16.26	13.29	11.16	9.56	8.32	7.34
1.00	98.00	48.05	31.40	23.08	18.10	14.78	12.41	10.65	9.28	8.17
1.50	146.7	71.90	46.92	34.45	27.00	22.00	18.50	15.86	13.80	12.17
2.00	195.0	95.30	62.0	45.40	35.50	28.90	24.20	20.70	17.99	15.84
2.50	242.9	118.0	76.70	55.97	43.60	35.40	29.59	25.22	21.86	19.20
3.00	290.0	140.5	90.90	66.10	51.30	41.60	34.60	30.08	25.43	22.30

TABLE II—(Continued)

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

	n (For K[1] interchange n and μ)									
μ	.11	.12	.13	.14	.15	.16	.17	.18	.19	.20
.01	.908	.916	.922	.927	.932	.937	.940	.944	.947	.948
.02	.816	.831	.844	.856	.865	.875	.880	.877	.893	.897
.03	.728	.748	.766	.783	.798	.810	.820	.829	.837	.845
.04	.634	.663	.688	.710	.729	.746	.759	.772	.784	.794
.05	.543	.580	.611	.638	.662	.683	.700	.716	.731	.743
.06	.452	.496	.534	.566	.594	.620	.641	.660	.677	.692
.07	.361	.414	.458	.495	.527	.557	.581	.604	.624	.641
.08	.270	.330	.381	.424	.461	.494	.522	.549	.571	.591
.09	.180	.247	.302	.356	.395	.431	.463	.492	.518	.540
.10	.0899	.165	.227	.282	.328	.869	.405	.436	.465	.490
.11	0	.0824	.152	.211	.262	.307	.347	.382	.412	.441
.12	.0899	0	.0758	.140	.196	.245	.288	.326	.360	.391
.13	.180	.0821	0	.0702	.131	.184	.230	.271	.308	.341
.14	.268	.164	.0756	0	.0654	.122	.172	.217	.256	.292
.15	.358	.245	.151	.070	0	.061	.115	.162	.205	.243
.16	.447	.327	.226	.140	.0652	0	.0573	.108	.153	.194
.17	.536	.406	.301	.209	.130	.0609	0	.0539	.102	.145
.18	.625	.489	.376	.279	.195	.121	.0571	0	.0509	.0965
.19	.712	.570	.451	.348	.259	.182	.114	.0538	0	.0482
.20	.800	.652	.525	.417	.324	.242	.171	.107	.0507	0
.30	1.67	1.45	1.26	1.10	.958	.836	.729	.633	.548	.472
.40	2.53	2.22	1.97	1.76	1.57	1.41	1.26	1.14	1.03	.927
.50	3.36	2.98	2.67	2.40	2.17	1.97	1.79	1.63	1.49	1.36
.60	4.18	3.73	3.36	3.03	2.76	2.51	2.29	2.11	1.94	1.78
.70	4.98	4.46	4.02	3.64	3.32	3.03	2.79	2.56	2.37	2.20
.80	5.76	5.17	4.67	4.24	3.87	3.54	3.27	3.01	2.79	2.58
.90	6.53	5.86	5.04	4.84	4.41	4.04	3.72	3.41	3.19	2.97
1.00	7.29	6.55	5.92	5.39	4.93	4.53	4.19	3.86	3.58	3.33
1.50	10.85	9.75	8.83	8.04	7.36	6.76	6.24	5.78	5.37	5.00
2.00	14.09	12.63	11.40	10.37	9.49	8.72	8.04	7.44	6.91	6.43
2.50	17.07	15.25	13.75	12.50	11.40	10.45	9.63	8.89	8.25	7.67
3.00	19.75	17.65	15.90	14.40	13.10	11.97	11.02	10.17	9.36	8.76

TABLE II—(Continued)

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

	n (For K^1 interchange n and μ)											
μ	.30	.40	.50	.60	.70	.80	.90	1.00	1.50	2.00	2.50	3.00
.01	.964	.972	.976	.978	.979	.980	.981	.980	.979	.976	.973	.969
.02	.928	.942	.951	.956	.959	.960	.961	.961	.959	.953	.945	.938
.03	.892	.915	.927	.933	.938	.940	.941	.942	.939	.930	.920	.909
.04	.857	.886	.903	.912	.918	.921	.922	.923	.919	.908	.895	.881
.05	.822	.858	.878	.890	.898	.902	.904	.905	.900	.887	.872	.856
.06	.786	.831	.855	.869	.878	.883	.886	.887	.881	.867	.850	.831
.07	.751	.803	.832	.848	.859	.865	.868	.869	.863	.847	.828	.808
.08	.717	.776	.808	.828	.839	.846	.851	.852	.846	.828	.807	.785
.09	.681	.748	.785	.809	.820	.829	.832	.835	.829	.810	.788	.764
.10	.647	.722	.763	.787	.802	.811	.816	.818	.812	.792	.769	.744
.11	.613	.695	.740	.768	.782	.793	.799	.802	.796	.775	.750	.725
.12	.579	.668	.717	.747	.765	.776	.783	.786	.780	.759	.733	.706
.13	.546	.642	.695	.727	.747	.759	.766	.770	.765	.743	.716	.689
.14	.512	.616	.673	.708	.729	.743	.750	.754	.750	.727	.700	.672
.15	.479	.590	.652	.688	.712	.726	.734	.739	.736	.712	.684	.656
.16	.445	.564	.630	.669	.694	.710	.719	.724	.721	.698	.669	.640
.17	.412	.538	.609	.651	.678	.694	.704	.709	.707	.683	.655	.625
.18	.380	.514	.588	.632	.660	.678	.688	.695	.693	.670	.640	.611
.19	.347	.488	.566	.614	.643	.662	.673	.681	.680	.656	.627	.597
.20	.315	.463	.545	.598	.627	.647	.659	.666	.667	.644	.614	.534
.30	0	.223	.348	.424	.472	.504	.525	.538	.552	.532	.503	.474
.40	.298	0	.167	.269	.335	.379	.409	.429	.458	.445	.420	.394
.50	.580	.209	0	.128	.212	.268	.306	.333	.381	.375	.356	.334
.60	.848	.403	.154	0	.1007	.169	.216	.250	.316	.318	.304	.286
.70	1.10	.587	.296	.117	0	.0801	.136	.176	.260	.271	.262	.247
.80	1.36	.758	.429	.225	.0916	0	.0646	.111	.212	.231	.227	.216
.90	1.57	.920	.552	.325	.175	.0727	0	.0527	.170	.197	.197	.189
1.00	1.79	1.07	.667	.417	.252	.139	.0585	0	.133	.167	.172	.167
1.50	2.76	1.72	1.14	.789	.555	.398	.284	.200	0	.063	.089	.0909
2.00	3.55	2.22	1.50	1.06	.774	.577	.437	.333	.111	0	.033	.0476
2.50	4.19	2.63	1.78	1.26	.935	.709	.547	.429	.178	.0417	0	.0196
3.00	4.74	2.96	2.00	1.43	1.06	.809	.631	.500	.182	.0714	.0425	0

TABLE IIa

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

Values of the efficiency of thrust bearings for screws, ($M \div M_s$), for various values of K^1 ($= 1 \div K$) as determined by Table II, and of A as determined by Formula 5. (Calculated by slide rule.) All efficiencies positive.

$$\text{Formula 5 is: } A = \frac{\mu_1}{\mu} \frac{t}{T} \frac{\rho}{r}$$

A (K^1 positive)									
K^1	1	2	3	.1 Diff.	K^1	1	2	3	.1 Diff.
100.0	1.01	1.02	1.03	.001	1.0	2.0	3.0	4.0	.1
50.0	1.02	1.04	1.06	.002	.9	2.11	3.22	4.33	.111
20.0	1.05	1.10	1.15	.005	.8	2.25	3.50	4.75	.125
15.0	1.066	1.133	1.20	.0067	.7	2.428	3.86	5.29	.143
10.0	1.1	1.2	1.3	.01	.6	2.66	4.33	6.0	.167
5.0	1.2	1.4	1.6	.02	.5	3.0	5.0	7.0	.20
4.5	1.22	1.44	1.66	.022	.4	3.5	6.0	8.5	.25
4.0	1.25	1.50	1.75	.025	.3	4.33	7.66	11.0	.334
3.5	1.286	1.572	1.858	.0286	.2	6.0	11.0	16.0	.50
3.0	1.333	1.666	2.0	.0333	.1	11.0	21.0	31.0	1.00
2.5	1.4	1.8	2.2	.04	.09	12.11	23.2	34.3	1.11
2.0	1.5	2.0	2.5	.05	.08	13.50	26.0	38.5	1.25
1.9	1.526	2.052	2.578	.0526	.07	15.28	29.6	43.9	1.43
1.8	1.556	2.111	2.667	.0555	.06	17.66	34.33	51.0	1.667
1.7	1.588	2.176	2.765	.0588	.05	21.0	41.0	61.0	2.00
1.6	1.625	2.25	2.875	.0625	.04	26.0	51.0	76.0	2.50
1.5	1.666	2.333	3.00	.0667	.03	34.3	67.6	101.0	3.33
1.4	1.714	2.428	3.143	.0714	.02	51.0	101.0	151.0	5.00
1.3	1.77	2.54	3.31	.077	.01	101.0	201.0	301.	10.00
1.2	1.833	2.66	3.5	.084	0	∞	∞	∞	
1.1	1.909	2.82	3.73	.091					

TABLE IIa—(Continued)

SCREWS GIVING WAY TO THRUST—LETTING WEIGHTS DOWN

All efficiencies above dark lines negative.

A (K^1 negative)

K^1	1	2	3	.1 Diff.	K^1	1	2	3	.1 Diff.
0					1.1	.091	.82	1.73	.091
0	∞	∞	∞		1.2	.167	.666	1.5	.084
.01	99.0	199.0	299.0	10.0	1.3	.23	.54	1.31	.077
.02	49.0	99.0	149.0	5.0	1.4	.286	.428	1.142	.0714
.03	32.3	65.6	99.0	3.33	1.5	.334	.333	1.0	.0667
.04	24.0	49.0	74.0	2.50	1.6	.375	.25	.875	.0625
.05	19.0	39.0	59.0	2.00	1.7	.412	.176	.765	.0588
.06	15.66	32.3	49.0	1.667	1.8	.444	.111	.667	.0555
.07	13.28	27.6	41.9	1.43	1.9	.474	.052	.578	.0526
.08	11.5	24.0	36.5	1.25	2.0	.50	0	.50	.050
.09	10.1	21.2	32.3	1.11	2.5	.60	.20	.20	.040
.10	9.0	19.0	29.0	1.00	3.0	.667	.333	0	.0333
.20	4.0	9.0	14.0	.50	3.5	.714	.428	.142	.0286
.30	2.33	5.66	9.0	.334	4.0	.75	.50	.25	.025
.4	1.5	4.0	6.5	.25	4.5	.778	.556	.334	.0222
.5	1.0	3.0	5.0	.20	5.0	.80	.60	.40	.02
.6	.666	2.33	4.0	.167	10.0	.90	.80	.70	.01
.7	.428	1.86	3.29	.143	15.0	.934	.867	.80	.0067
.8	.259	1.50	2.75	.125	20.0	.95	.90	.85	.005
.9	.111	1.22	2.33	.111	50.0	.98	.96	.94	.002
1.0	0	1.0	2.0	.10	100.0	.99	.98	.97	.001

TABLE III

SCREWS OVERCOMING THRUST

Maximum efficiencies for various sets of values of slopes of threads, n, and coefficient of friction, μ.*

Coefficient of friction η	Slope of threads for maximum ratio n	$(M_t \div M_s)_m$ Maximum ratio or efficiency
.01	.990	.980
.02	.980	.960
.03	.970	.942
.04	.961	.924
.05	.951	.905
.06	.942	.887
.07	.932	.870
.08	.923	.852
.09	.914	.836
.10	.905	.820
.11	.896	.803
.12	.887	.786
.13	.878	.772
.14	.870	.756
.15	.861	.741
.16	.853	.727
.17	.844	.718
.18	.836	.699
.19	.828	.686
.20	.820	.672
.30	.744	.554
.40	.677	.458
.50	.618	.382

*The reason why the efficiencies for a given value of the coefficient of friction are the same in Tables III and IV is because we have the following identity from trigonometry:

$$\frac{1}{\tan\left[\frac{\pi}{4} - \frac{\phi}{2}\right]} = \tan\left[\frac{\pi}{4} + \frac{\phi}{2}\right] \qquad \text{(See Equation 8).}$$

TABLE IV

SCREWS GIVING WAY TO THRUST

Maximum efficiencies for various sets of values of slope of threads, n, and coefficient of friction, μ.*

Coefficient of friction μ	Slope of threads for maximum ratio n	$(M_s \div M_t)_m$ Maximum ratio or efficiency
.01	1.010	.980
.02	1.020	.960
.03	1.030	.942
.04	1.041	.924
.05	1.051	.905
.06	1.062	.887
.07	1.072	.870
.08	1.083	.852
.09	1.094	.836
.10	1.105	.820
.11	1.116	.803
.12	1.127	.786
.13	1.138	.772
.14	1.150	.756
.15	1.161	.741
.16	1.173	.727
.17	1.184	.713
.18	1.196	.699
.19	1.208	.686
.20	1.220	.672
.30	1.344	.554
.40	1.477	.458
.50	1.618	.382

*See note, Table III.

TABLE V

Illustrating the effect upon the efficiency of a screw caused by changing the pitch while all other dimensions remain unaltered. The screw turns and lifts a weight, the fixed dimensions and working conditions having been described on page 393.

Tan θ slope of thread	Efficiency of pivot	Efficiency of threads	Net efficiency	
n	$M_s \div M$	$M_t \div M_s$	$M_t \div M$	Remarks
.04	.779	.284	.221	
.05	.790	.331	.261	
.06	.802	.373	.299	
.07	.810	.410	.332	
.08	.820	.442	.362	
.10	.835	.495	.413	
.125	.850	.550	.468	
.50	.941	.791	.744	
.905		.820		(Maximum value thread efficiency.)
1.05	.969	.817	.792	(Maximum value net efficiency.)
1.50	.98	.796	.780	

TABLE Va

Illustrating the effect of a hand-lever upon the efficiencies of the screws given in Table V. The length of the lever is eight times the average radius of the screw-threads, and the radius of the journal is equal to the average radius of the threads.

n	$(M_t \div M_L)$
Slope of threads	Net efficiency
.04	.219
.05	.258
.06	.296
.07	.329
.08	.358
.10	.409
.125	.463
.5	.737
1.05	.784
1.5	.772

TABLE VI

Illustrating the effect upon the efficiency of a screw caused by changing the pitch while all other dimensions remain unaltered. The nut turns while the weight settles, the negative net efficiencies indicating that the slope of the thread is not sufficient to admit of the settling of the weight without the aid of an assisting torque upon the nut. Dimensions and working conditions given in the text.

n	$M \div M_s$	$M_s \div M_t$	$M \div M_t$	
Slope of threads	Efficiency of pivot	Efficiency of threads	Net efficiency	Remarks
.04	+3.014	—1.492	—4.50	
.05	+3.410	— .995	—3.395	
.06	+4.009	— .665	—2.663	
.07	+4.991	— .429	—2.14	
.08	+7.059	— .248	—1.75	
.10	$\pm\ \infty$	$\mp$ 0	—1.2	(Net efficiency = A when $(-\phi) = \theta$)
.125	—3.860	+ .1975	— .7625	
.200	— .225	+ .490	— .110	
.223	$\mp$ 0	+ .539	$\mp$ 0	(Screw locks for negative values of net efficiency)
.3	+ .381	+ .647	+ .247	
.4	+ .582	+ .720	+ .419	
.5	+ .687	+ .765	+ .525	
1.0	+ .853	+ .817	+ .697	
1.105	+ .867	+ .818	+ .709	(Maximum thread efficiency)
1.5	+ .901	+ .811	+ .731	
1.74	+ .914	+ .803	+ .734	(Maximum net efficiency)
2.0	+ .924	+ .792	+ .732	

TABLE VIa

Illustrating the effect of a hand-lever upon the efficiencies of the screws given in Table VI. The length of the lever is eight times the average radius of the threads, and the radius of the journal four times the average radius of the threads. The nut turns and lets a weight down.

n	$M_L \div M_t$
04	—4.59
.05	—3.46
.06	—2.72
.07	—2.18
.08	—1.78
.10	—1.22
.125	— .778
.20	— .112
.223	0
.3	+ .242
.4	+ .411
.5	+ .514
1.0	+ .683
1.105	+ .695
1.5	+ .716
1.74	+ .719
2.0	+ .718

TABLE VII

VALUES OF SEC α FOR VARIOUS SETS OF VALUES OF n AND β

Half angle of thread	n = Slope of thread at point of application of resultant				
β	0.176	0.364	0.577	0.839	1.0
10°	1.015	1.014	1.012	1.009	1.007
20°	1.062	1.056	1.048	1.038	1.033
30°	1.15	1.138	1.118	1.094	1.084
40°	1.297	1.275	1.235	1.19	1.168
45°	1.404	1.373	1.323	1.26	1.231

THE TRANSIT COMMISSION

By E. K. Morse*

In discussing the work done by the Transit Commission of the City of Pittsburgh it is proper to consider first the steps followed by Council in creating this Commission; especially as it has to deal largely with the relation of the Engineers' Society of Western Pennsylvania to the city administration.

In the first place, Mayor Armstrong requested that Mr. Samuel E. Duff, president of the Society, should appoint a committee to consider the question of congestion and its relief in the down-town section. This committee was appointed and reported back to the Society. Later on, Council called for a public meeting to consider this report and many of our leading engineers took part in the open discussion in Council chamber. The result of this discussion, and the recommendations furnished by the members of our Society, led Council to prepare and pass an ordinance creating the Transit Commission, which was signed by the Mayor on Nov. 13, 1916. On the same date a recommendation was passed calling for a legal investigation of the street-car service. It was the author's good fortune to be appointed Transit Commissioner.

It was no easy matter to obtain men skilled in transit matters and to complete an organization adequate and sufficient to deal with a subject of this magnitude, especially during these war times, but the author has been successful in surrounding himself with a splendid organization—the best that he has ever had.

In 1910 Mr. Bion J. Arnold reported to Mayor Magee on the condition of the service furnished by the Pittsburgh Railways Company, but although he published an exhaustive report, he entirely failed to give a single clear, definite recommendation, and little or nothing has been obtained from that report other than compilations of valuable data. In the study of the present problem the statistics for this city have been completed as they pertain

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to traffic, from 1910 to 1917, in such a way that the records are complete, from the beginning of Mr. Arnold's work, to the year 1918.

The Commission was created on account of congestion in the Central Business District, on account of inadequate street-car service, and to report on the subject of rapid transit. One of the first objects desired was to obtain from the officials of the Pittsburgh Railways Company all the immense amount of data they had in their possession, but this was promptly refused by the Railways Company and, in fact, all friendly co-operation denied. The refusal to give data was based upon the fact that they were being prosecuted by the Legal Department of the City of Pittsburgh. The author can excuse them for denying the Transit Commissioner the privilege of copying all their records but he cannot forgive them for refusing a schedule, the privilege of measuring up the size of the cars, or the privilege of stepping into their yards, etc. The denial of such co-operation reveals a small spirit and one that will, if continuously indulged in, bring financial ruin to the Company, as no service company can antagonize the very people it proposes to do business with, and have the co-operation that is essential. But we will come back to this subject later.

When the Pittsburgh Railways Company refused all co-operation, Council was asked for and granted $10 000 additional money for the survey of travel—in other words, to enable the Transit Commission to count all the people that came into the city, and all the people that passed out of the city during business hours daily. An additional organization was created and the most hearty co-operation of the University of Pittsburgh and the Carnegie Institute of Technology made it possible to have on the payroll 315 students from these two institutions — bright young men, anxious to earn a little money and willing to please—thus making it possible during this critical labor market to complete as fine a lot of surveys of the traveling public as any city has ever made; and for one-third the amount of money that has ever been expended for the same amount of data. The City of Pittsburgh has in this one item saved not less than $25 000 over

and above the cheapest survey of its nature and magnitude ever made before.

Men placed on the cars at the city's expense rode backwards and forwards from five o'clock in the morning until midnight, counting the number of people that got on and off all the cars that came into the city and that went out, on all routes. Pedestrians were counted, and people who rode in automobiles; people were counted and checked on the steam railroads; people were counted and checked as they crossed backwards and forwards on the bridges. In fact, traffic by every method of communication was calculated. The managers of the steam railroads furnished the most hearty co-operation. By compiling all counts made, it was found that on an average day 282 253 people entered the Central Business District. Of this average, 165 452 came on the street-cars, 30 930 on the steam railroads, 37 805 by automobile, and 47 816 on foot. Do not fail to note the significance of the last figure.

On July 2, 1917 a count was made of the vehicular travel from 7 a.m. to 7 p.m. in the Central Business District, which has been defined as that area bounded by the P. C. C. & St. L. R. R., Eleventh Street and the Allegheny and Monongahela Rivers—an area less than 250 acres. The total number of vehicles of all classes entering this district was found to be 24 904 and those passing out, 24 917. The curve of this traffic shows that some 2400 vehicles, mostly pleasure automobiles, remained in the city all day. Some of these machines remained on such streets as Wood Street and Fourth Avenue as long as 7½ hours without moving; notwithstanding that a city ordinance says that one-half hour shall be the limit of time. During the same day that the vehicle count was made, 5301 street-cars entered the same district, making a grand total of 30 218 vehicles and street-cars that entered and passed out of the down-town district. In addition to this the taxicabs passing backwards and forwards through the city, and the horse drawn vehicles, trucks, delivery wagons and automobiles that did not pass beyond the limits of the throats entering the city, were not counted. It is believed that this cross-town movement was equal to about one-third of the actual count, and that the grand total was easily 40 000 vehicles and street-

cars passing through our narrow congested streets. Think of it! A total of 40 000 vehicles and street-cars on the surface in the Central Business District—a business area less than one-half that of any business area in the world which compares with it in volume of business transacted per square foot of surface.

The most comprehensive and the most interesting count was that of the Industrial Survey which was made to determine where the workers lived and where they worked; and from that survey to try to arrive at conclusions as to where they should work and where they should live, what kind of service should be given and what kind of houses should be provided in order to give them attractive homes. Something to work for, something to long for; and when once obtained in the shape of a nice bungalow home with sufficient garden space to raise their own vegetables—then would be appreciated what it means to have a contented labor market, not only with common labor but with the very highest skilled labor. The problem is, will Pittsburgh arise to the occasion? Will the present Mayor and Council make good? If they are dependent upon the loyalty of the Engineers' Society of Western Pennsylvania, and the individual support of the engineers, then they will make good.

The Transit Commission studied exhaustively the question of population of Pittsburgh and vicinity up to and inclusive of 1916. In this study it was shown that the population of Pittsburgh was 596 000 in 1916—a growth of about 11.6 per cent. since 1910. The Metropolitan District—that is, Greater Pittsburgh—was estimated to contain 1 175 000 persons in 1916—an increase of 16.8 per cent. In making the Industrial Survey, the total population of Pittsburgh and the surrounding district was taken at 1 160 000. The assumption was made that one person in every three was an industrial worker or student in the advanced schools and the total number was assumed at 386 700. The check reached and located 274 000 persons in all, or 70 per cent. of this total, but it is believed that this 70 per cent. will more properly represent the workers in the manufacturing districts. This count was made from payrolls, of men and women actually at work, furnished by 796 firms and institutions. The

author has time merely to touch upon this most interesting and valuable survey.

The trip survey showed that the average ride of a passenger on the street cars was 3.0 miles, while that shown by the Industrial Survey was only 2.52 miles as against 3.5 miles claimed by the Pittsburgh Railways Company. It further showed that a certain class of labor came from certain sections of the city and that in the slums of our city 65 000 people out of these 274 000 lived in hovels alongside of the manufacturing and industrial plants and that something like 35 000 of them walked. By studying the maps made from this Industrial Survey, after they were most carefully compiled, the necessity for short, balanced, through routes was arrived at and the demand for additional shuttles and the need for a surface belt-line around the city were clearly brought out. It also emphasized the necessity of turning back long routes as quickly as possible—turning them back at the very edge of the Central Business District and transferring either to the belt line or to the shuttles, all those who wanted to continue their ride.

The study of the flow of traffic and its comparison with other cities brought to light one condition that the Commissioner feels is vital and one that must be corrected or the Pittsburgh Railways Company must go into the hands of a receiver. That is, the riding habit or rather the lack of riding habit in the City of Pittsburgh. Mr. Arnold's report showed the riding habit of the City of Pittsburgh in 1910 as 204 rides per person per year. When the Pittsburgh Railways Company refused co-operation there was nothing for the Transit Commissioner to do but to take the sworn statement filed by the Pittsburgh Railways Company with the Pennsylvania Department of Internal Affairs at Harrisburg—for purposes of taxation—which gave the riding habit of Pittsburgh in 1916 as only 217. At the same time Boston's riding habit was 313, or 44.2 per cent. more than that in the City of Pittsburgh. Chicago was 341, or 57.1 per cent. more than that in the City of Pittsburgh, while the riding habit in 1916 in Detroit had increased from 275 in 1910 to 362 in 1916 —a figure 66.8 per cent. greater than that in the City of Pittsburgh. The passenger revenue in 1916, taken from the sworn

statement of the Pittsburgh Railways Company was $12 500 000. Had 67 per cent. more people ridden, or the same as rode in Detroit according to the riding habit, they would have increased their revenue from passengers $8 375 000, or a total of $20 875 000 instead of $12 500 000. Using the riding habit record in Boston, the increased revenue would have been $5 525 000, making a total of $18 025 000 instead of $12 500 000. Boston is a fairer comparison than Detroit. The author has already stated that the riding habit increases in direct proportion to the facilities furnished. Furnish the best of service and you will have the highest riding habit. Accommodate the public—give them splendid service, splendid cars and courteous treatment—and the riding habit will increase directly with the accommodations and facilities furnished. It may be further noted that in those cities where there is a large riding habit there is a practically universal transfer privilege. The transfers in Pittsburgh are only about 12 per cent. in comparison with from 40 to 50 per cent. in other cities. The Pittsburgh Railways Company has systematically refused and ignored the constant appeal for more transfers, the constant request for better service and the constant demand for more cars. It is believed that the traveling habit instead of going up and reaching close to the 300 mark will go below the 200 mark, and if it does it spells bankruptcy. The traveling habit in this city should not be, under any condition, less than 275. It should be more—it should be 300—and it would be if the service demanded were granted.

It is claimed that, on account of the rough, rugged topography and the grades that are on all lines, it costs more for operation. To be exact, the author will quote from the language given out by the Pittsburgh Railways Company's folder (page 9) as follows, "The five-cent fare in hilly Pittsburgh is only one-half as much per passenger mile as the five-cent fare in most level cities." Let us turn to page 23 of the Report of Transit Commissioner and look at the gross revenue earned in Chicago which is the one city that has furnished complete records for a series of years. The gross revenue in Chicago in 1913 was 29.58 cents per car-mile, while that of Pittsburgh in 1913 was 30.74 cents per car-mile and in 1916 was 33.88 cents per car-mile. The

operating cost in Chicago was 17.4 cents in 1913 while that of Pittsburgh was 20.24 cents and, in 1916, 20.39 cents. The net earnings in Chicago in 1913 were 12.12 cents and in Pittsburgh for the same year 10.50 cents, and in 1916 they had risen to the very high figure of 13.49 cents. If it costs more to operate in the City of Pittsburgh, then why should the power per car-mile in Chicago in 1913 amount to 2.77 cents while that of Pittsburgh was only 3.37 cents for the same year and, in 1916, 3.23 cents per car-mile? You will note that these comparisons are very favorable and, if you will consult the table on page 23 of the Report of Transit Commissioner, you will find that all the other figures given there compare favorably.

Now, for a moment, let us see what was done with this revenue. In Chicago the total return on operating properties—that is, the net receipts—was 8.99 cents in 1913. The Company's share and interest showed 6.78 per cent. In addition to this the operating company in Chicago paid over to the city as its share of the profits of operation the sum of $2 698 965.42 as shown on page 21 of the Report. The same year Pittsburgh had a deficit of $473 644. The leak which is due to financial overburden—which no management can carry successfully—is the great factor that is bringing about a handicap to the proper growth of this city, and that must be corrected, or else the City of Pittsburgh in a very few years will occupy, instead of third place, fifth or sixth place in her industrial growth, and the homes for mechanics—the bungalow homes—cannot be located as they should be, without rapid transit.

While the different surveys were being carried on, a very careful study was given to the subject of city planning and, as it has been my privilege to be a member of the City Planning Commission for eight years, this privilege furnished an opportunity to put in print recommendations that had been gradually studied to a conclusion. Without going into the topography of the city, the rivers form the city into one great wedge, and all the traveling public of every description coming in through the different throats into the Central Business District only wedge themselves closer together as they come down into the center of the city, and finally become so packed that the congestion is, even

to Pittsburghers, unbearable. There must be immediate relief or else the city must either grow towards East End or, preferably, over on the North Side where the growth always should have been, especially as there are three acres of ground available on the beautiful plateau on the North Side to every one in the Central Business District.

In the study of the street problem it was found by comparison with other cities that the street area was from 30 to 40 per cent. less than that of almost any other city of the same magnitude, and that to compare favorably with other cities no street in the down-town Central Business District—the "Golden Triangle"—should be less than 60 feet, and from that width up to 80 feet, whereas but one street—Liberty Avenue—is 80 feet in width. This condition of affairs led the Commissioner to recommend first, in the solution of surface travel and rapid transit, a program of city planning.

The first and most important street problem to be dealt with is the widening of Second Avenue from 40 feet to 80 feet, from Grant Street to Liberty Avenue. This is the key to the situation. The widening of this street will bring into the city the whole Point District, which is now in a financial slump. The widening of this street will make a boulevard which eventually will be a continuation of the Monongahela Boulevard and, in order to complete the triangle, Grant Street should be widened to 80 feet, and the Lincoln Highway thrown around the city and across the new Manchester Bridge at the Point to the North Side —a by-pass as it were. In addition to the widening of Second Avenue and Grant Street, Water Street and Duquesne Way should be widened in order to give a 48-foot driveway around the city. West Carson and Diamond Streets should be widened according to the plans now made by the City of Pittsburgh. Some of this work should be done now and in other cases the ordinances should be passed and plans made in order that they will be ready to proceed with immediately after the war is over. If Second Avenue and Diamond Street are widened, a complete system of rerouting can be brought about that will furnish the traveling public with a belt-line system running both ways through the whole Point District and around and through the

Central Business District; a shuttle run from the P. & L. E. R. R. depot to the P. R. R. depot and the continuation of the shuttle now running from the Court House to the North Side; and short, balanced through routes through residential, industrial and mill sections of the whole city, with transfers to each through route, the shuttles, the belt-line, the other short through routes and to the inclines. This latter, balanced, through routing will take care of the laboring classes and our mechanics in a way that is not now provided and will increase the riding habit, as it has never before increased. It will reach the 47 000 people who walk into the city and who will find it to their advantage to ride and accept a transfer. The records of the Pittsburgh Railways Company show conclusively that there are big earnings in the nickel if the burdens carried were not impossible. The riding habit must be brought up to at least 275. If it goes on down at the rate it is now slumping it will be a case of receivers for receivers.

We have now led up to the subject of rapid transit—a subject which the author would have liked to discuss at this time, but which would take up too much time to enter into and properly handle in a lantern-slide talk, but which he will be pleased to present to you, if you wish, at some later date.

DISCUSSION

Mr. Samuel E. Duff:* When I first saw the Report of the Transit Commissioner it struck me that he had made a very logical arrangement of his recommendations. It has been the habit of everybody when talking about congestion in down-town Pittsburgh to consider that it consists only of crowded and delayed street-cars, and to forget all about the pedestrian and vehicular congestion. I think Mr. Morse is very wise to deliberately reverse the usual order.

Let us hope that engineering treatment of this question of street congestion outside of street-car transportation or rapid transit may lead to action on some logical system for rebuilding or replanning down-town Pittsburgh. Our great trouble with anything of that kind is that everybody who happens to have property liable to be touched by the march of progress demands exorbitant damages and is enabled to hold up improvements until popular interest declines. The rights of individuals have been so bulwarked with all kinds of laws, ancient and modern, without any regard whatever for public exigency, that the necessary remodeling of congested city areas is almost impossible.

Look at the map of the ordinary city such as Mr. Morse has shown us, and we see a lot of rectangular blocks and streets of approximately equal width without any regard whatever for traffic requirements. Everybody who has ever observed a city knows that traffic is not uniformly distributed over any section. Are we going to sit around here—we and our children after us—and have our business constricted and made more expensive simply because a lot of fellows 125 years ago laid out these streets? I think it is about time that we straighten the situation out. Mr. Morse has suggested a number of things that ought to be done and I have no doubt he could suggest a lot more. But the sentiment in this city against street widening, or any interference

*Consulting Engineer, Pittsburgh.

with the city planned 125 years ago, is so great that I have no doubt he hesitated to suggest a lot of other things he knows should be done.

If the engineers of Pittsburgh and the members of this Society think about this and talk about it long enough and strongly enough the community will wake up and believe that they should pay less attention to lawyers, and laws made by lawyers for the purpose of increasing the activity and fees of lawyers, and go after the thing along common-sense lines as dictated by engineering research and with proper regard for the rights of the public. The trouble is that the fellows who own property and control the whole thing are traders, in the sense that they want to get for themselves all there is in it. That is all right, but the rest of us are suffering and somebody has to point out where the benefit to the community lies and some large group of men should insist that the subject be debated until the community sees it in the right light.

We are glad to have Mr. Morse tell us about the Industrial Survey he made, and explain the technical details of it, because I think it has never been carried out to the same extent anywhere else. I do not think the methods used in the surveys made in Philadelphia and other cities were quite so far advanced as the methods Mr. Morse has used.

Mr. W. E. Snyder:* In considering the changing and widening of streets for the purpose of eliminating the congestion which now exists in Pittsburgh, it is well to compare the conditions which now exist in Boston. That city was founded probably 130 years before Pittsburgh was. Many of its streets are narrow and crooked, and, so far as can be judged now, the time has gone by when they can be changed and the thoroughfares of the city modernized. This is not yet the case in Pittsburgh, but it will be just as soon as the streets are built up with large, modern office buildings or warehouses.

It may be of interest to mention briefly changes made in one of the older cities in Germany, the city of Metz. I stayed there

*Mechanical Engineer, American Steel & Wire Co., Pittsburgh.

in 1908 sufficiently long to get well acquainted with the conditions. In the older part it had some features which reminded one of medieval times. The streets were narrow and crooked, with little or no sidewalk; buildings quaint and old-fashioned and the town surrounded by earthwork fortifications. I was there again in 1913, and a surprising change had taken place in the interim. The earthworks were replaced by nice parks and drives, and some of the streets were improved by straightening and widening, though not in the actual business center. It has undoubtedly been the experience of many European cities, as well as the older cities of this country, that advantage was not taken of opportunities to straighten and widen some thoroughfares and thus eliminate congestion before the boundaries of these streets were permanently fixed by large modern buildings.

MR. L. P. BLUM:* There is no other question so difficult to handle in the older municipalities as that of widening streets, and getting adequate width for the traffic. Traffic has grown beyond the dreams of the men who planned the streets. It is very easy to talk about widening streets but if we attempt to do even half justice to the adjoining property we will find the cost something tremendous. As illustrating that fact I have made some studies of this central Pittsburgh district and there have been only about three street widenings in what was the original Colonel Wood Plan of Pittsburgh. In Philadelphia, there have been only about three, the matter of expense being the chief consideration in each case.

There are, however, several things which it seems to me ought to be adopted in Pittsburgh, and of which we have, perhaps, examples in the neighboring city of Philadelphia. Mr. Morse has shown the arcade system for street corners—a most excellent idea which could be carried further, to the actual arcade treatment of whole blocks of city streets; as in the case of Fifteenth Street between Market and Filbert in Philadelphia where the Pennsylvania Railroad, owning the adjoining property, has the sidewalk entirely under cover. This might be a proper solu-

*Blum, Weldin & Co., Pittsburgh.

tion for Grant Street. I was glad Mr. Morse emphasized the matter of Second Avenue. I think the city should carefully consider the advisability of widening this street where property values are comparatively low.

Another thing that strikes me as peculiar is the different municipal laws of Pittsburgh and Philadelphia, for instance. In Philadelphia the city has the power to straighten streets in advance of construction. For instance, I worked in a certain suburban district in Philadelphia which was largely farm land. Streets through these farms had been projected for twenty years and no one could build even a farm-house covering any portion of a proposed street, but the city need not pay damages until the actual opening of the street. I remember a few years ago there was the attempt to apply the same principle in Pittsburgh and the Supreme Court held that the city must open the street physically and pay damages as soon as the street was planned. I think it would be very proper to have the Philadelphia plan applied in Pittsburgh.

The Report which Mr. Morse has published ought to be appreciated. I would like to ask a few questions with reference to it. On page 9 Mr. Morse speaks of the proportion of street to building area comparing Pittsburgh with other cities. I lived in Philadelphia for some years and it certainly seems to me that the proportion of street to building area in Philadelphia is less than that in Pittsburgh. Perhaps Mr. Morse's proportion comes from including in the building area the large wharf area of which he spoke. Was that included?

Mr. E. K. Morse: No.

Mr. L. P. Blum: I am also impressed in this study by the fact that the maximum increase of population has been in those districts just within the limits of the five-cent fare. As we come closer to town, and as we go beyond the five-cent limit, we find that the rate of growth has not been so great. That means simply what real-estate men have told me many times—that people have a certain reluctance to paying more than a five-cent fare and it shows the necessity for keeping the limit of the five-cent fare at least as widely extended as at the present time.

One is very much impressed by the figures shown on page 28 of Mr. Morse's report in which he shows that the average haul per passenger in the City of Pittsburgh is not over three miles per carfare. One who does not go into figures would suppose it to be more nearly five miles, but I have no doubt that if Mr. Morse included the North Side lines the figures would be even less than three miles.

Regarding this question of congestion, not only present but future congestion must be considered. To-day I happened to see the results of traffic censuses in Chicago last year and 10 years ago. In 10 years the automobile traffic has become 10 times as great, while the entire vehicular traffic has doubled. Doubtless the same figures apply to Pittsburgh. And if the next 10 years will show a doubling of vehicular traffic we can see what this congestion of Pittsburgh streets is going to amount to.

MR. EDWARD GODFREY:* In 1913, before the Hungry Club at the Fort Pitt Hotel, the speaker made an address in which he outlined a solution of Pittsburgh's traction problem. The address was very fully reported in the *Pittsburg Dispatch*, of Dec. 30, 1913. The plan contemplated the following:

1. Elevated loops in the down-town portion of the city from the "Hump" and back.

2. Elevated tracks to the North Side, reaching about to Ohio Street.

3. Elevated tracks to the South Side.

4. An elevated line out Liberty Street to about Main Street.

5. An elevated track or viaduct spanning the dip in Soho and coming to the surface toward the top of Soho Hill.

6. The use of Forbes Street only, for street-cars as far as Oakland, leaving Fifth Avenue for automobiles and teams.

7. A routing that would provide cars at close intervals going out Forbes Street to Oakland, then out Fifth Avenue to Point Breeze, then in Penn Avenue to town, other cars running at the same time in the other direction on this same route.

8. Transfer stations where passengers could step off the

*Structural Engineer, Pittsburgh.

through cars and await, under shelter, the branch car desired. These stations would be located in town where the South Side and North Side cars branch off, at Soho, at Atwood Street, at Point Breeze, and at other points where branches join this main route.

9. Loop cars to all of these stations serving the branch lines.

The advantages of this plan are many. Among them are the following:

1. All cars leaving town could be uniformly loaded, for every car would go exactly the same way and any car would take any passenger at least a part of his way. There would be no such thing as two or three hundred people waiting in town for a car on a certain line that traverses the outlying districts of the city and may have all of its cars held up an hour out in the suburbs.

2. Cars could travel rapidly, for in the down-town section the tracks would be perfectly clear; out Liberty Street they would be the same; and with teams and automobiles diverted to Fifth Avenue as far as Soho, there would be little interference.

3. Stops at wide intervals along the route would then be rational, for some speed could be attained between the stops. It is impossible to see the benefit to any one occasioned by the stops at wide intervals in Pittsburgh's present traction system, particularly in the down-town district. There may be some way to make it clear to an imbecile that a car which takes five minutes to travel two long blocks, stopping of necessity for half of that time and refusing in that period to open the doors at corners where the cars invariably stop, is thereby improving the system.

4. The "first car", "second car", "third car" abomination would forever be eliminated or have its sting withdrawn. At present, if you are not standing at the exact spot where the conductor thinks you ought to be standing, or if someone else who is standing in front of you does not want that car, snap goes the door, and you may have to wait 30 to 60 minutes for another car on your line.

5. There would be no dumping of a carload of people on the street in all kinds of weather, with the inexplicable "Take the

next car", for all necessary changes could be made at the sheltered stations. Transfer points would not be pneumonia stations.

6. This system would not do violence to property values that have grown up around the present location of street-car lines in the down-town streets, where large buildings have their property values vitally affected by the proximity of street-car lines.

7. This system would bring passengers to the business center of Pittsburgh. As Mr. E. K. Morse said in the PROCEEDINGS of the Engineers' Society of Western Pennsylvania, March 1907, "If the East End line should bring their passengers only to the Court House, they would satisfy nobody." Mr. Morse also advocated terminal loops, and the delivery of passengers to as many points as possible and as near their objective points as possible—both of which this system would accomplish.

8. This system would not entail the widening of streets at great expense to the city.

9. It could be carried on piecemeal with very little inconvenience and with scarcely any interference with present street-car traffic.

The public is beginning to realize the fact that a ten-story city or portion of a city needs more than a one-story street.

Even the Public Service Commission is beginning to see the folly of long car routes with branches leading off away out in the outskirts of the city—branches on which every car must traverse the entire route. It is impossible both in theory and practice, to keep any kind of schedule time in a system of this sort. Only this morning we are told that the Public Service Commission is seeing a light and has directed, or requested, that the three lines that go out Butler Street to Sixty-second Street, and then spread out, be replaced by one line to Sixty-second Street and two feeders. It is exactly this, on a large scale, that the speaker advocated in 1913.

Elevated roads can be built with their posts in the curb line where the present trolley posts are placed. They can be made ornamental by covering the girders with concrete. Noise can be reduced by using ballast and a reinforced concrete slab. All of these things are being done now in reconstruction work in

Brooklyn. Thus the stock objections to elevated tracks are nullified.

Subways for Pittsburgh would mean unprofitable long hauls, trouble from floods, and 10 or 15 cents carfare for residents of Pittsburgh to reach their homes. Subways are scarcely to be thought of in a city of the topography of this one.

A short time ago, in the *Pittsburg Dispatch*, the speaker published a suggestion which would give unbounded relief to Pittsburgh's traffic congestion. It could be put into operation as a portion of his inclusive scheme, or it could be done as a thing by itself. The suggestion is to make a dip in Webster Avenue or the Boulevard, where it crosses Sixth Avenue. The greater number of the automobiles and street-cars that enter and leave the city, pass this intersection. If they passed on different levels, all interference would be avoided and congestion that at times reaches for many squares each way would be eliminated. A block or so each way would be enough to give the desired grade, and during the change the boulevard could be carried on timber construction, thus affording almost immediate relief.

Mr. Lee C. Moore:* There are numerous solutions, but what Pittsburgh wants is action. I might say right now that I have solved the problem from a selfish point of view. I am going to move away and establish myself in a western city.

I was much interested in Mr. Morse's reference to the city docks, and the comparison of our docks with foreign docks. I think he is right, and though I never thought of it until he brought it out, I see no reason why we should have this dead triangle down in the business section. It keeps spreading and by and by it will spread over and wipe out Fifth Avenue and some more of the town. Why it is dead we all know.

Mr. W. E. Snyder: The foregoing discussion reminds me of a question I would like to ask. Undoubtedly this is a splendid report, and some of the recommendations made in the report are capable of being carried out in the course of the next two or

*President, Lee C. Moore & Co., Pittsburgh.

three years, if definite efforts are made to do so. Some of the recommendations, in fact, could be carried out in less than three years. Now the question is, what is the next move in a case of this kind? Whose business is it to take up and carry out these recommendations, which are comparatively easy to put in effect. Is it a matter of follow up by some person or persons to persuade some other persons, principally engaged in building political fences, that this report contains a number of recommendations, which, if carried out, will be of great benefit to Pittsburgh?

Mr. E. K. Morse: I will be glad to answer that question. There will be no result if the Report is left to itself. It will be covered with dust and when the city gets ready to do something to the traction lines there will be another report. That is history. That will happen this time unless some one will get busy. You have no right to expect the present Public Service Commission to revolutionize things. But there is this present opportunity, and that is the reason this subject was split into two parts, the first part containing recommendations that can be put into operation now. Mr. Sprague will tell you that it will take two years to open Second Avenue, get the bills ready, get the law department to move, get the condemnation proceedings, and things of that sort; but there is nothing to prevent paving Water Street to-day if you want to.

The remark was made that the cost of widening streets is nearly prohibitive. From figures that have been given me, the city taxes on this building and the one across the way—the William Penn Hotel—are paying the interest on the whole cost of widening and grading the "Hump," and will retire the bonds in 10 years before maturity. You could not ask for a better investment than that. And those are city figures, not mine. There is no use of talking about subways and elevated roads now, during the present war, but there is every reason to decide now what you are going to do, and to be prepared; just as Detroit is now getting ready to build. Why? Because she is just simply going to skin Pittsburgh of her labor market and her industries. It is a shame that we have to ship raw material to Detroit and stand on the corner while trucks pass out the Lincoln Highway carrying the

finished products which should be and could just as well be fabricated here. There is no reason why the whole district out Penn Avenue to Sixty-second Street should not be denuded of the hovels that cover it, and replaced by diversified industries.

In all the deductions we have made we have tried to base our calculations on revenue to the operating company. We have tried in our deductions to adopt nothing that was not a revenue producer for the Pittsburgh Railways Company, believing that you can not have good service from a company that is not making money.

But, to further clench your suggestion, Mr. President, if you depend on the Transit Commission alone, you will get mighty little done, while if you will join in and we all pull together results may be expected.

Mr. Harry J. Lewis:* The present period is one of changing view-point with regard to the natural monopolies covered by the term "public service." This service has been largely taken over by corporations and operated with the intention of yielding the highest possible income on capital. The quality of service has been a secondary matter, in many cases, and the neglect of this point has almost always led to financial losses. Lately, however, the public has come to a dim realization that it furnishes all of the revenue and is thereby the principal party in the business, so that quality of service and the price charged are the real issues to be determined. The courts are backward in this as in many other things and have not yet caught up with the public mind.

If a public service corporation ruins its business and its income with poor service and excessive charges, it would seem but just that it should share the fate of any other enterprise which goes to the wall for the same reasons.

No private corporation is going to furnish a high-grade service without a fair reward and it may be that the present situation will develop a condition of insolvency in our present companies which they will be forced to admit. The city will then have a chance to bid in the property in competition with private capital

*Consulting Engineer, Pittsburgh.

and to eliminate much of the artificial value which is the basis of present contention.

Our present rate of fare and the quality of service are drawbacks to the whole community, and account to some extent for our sluggish increase in population as compared with Detroit, for instance. The greater increase in the riding habit in Detroit, over Pittsburgh, is probably very directly connected with the fact that Detroit has a lower rate of fare and a better quality of service. Increase in the riding habit almost invariably lowers the average length of ride as the long-haul patrons are most of them bound to ride anyway.

The public is willing to pay all of the real cost of haul plus a reasonable profit, as it will have to do either under public or private ownership, but it is mustering a very stiff resistance against paying a return on artificial values or capitalization. Until these things are to some extent settled and recognized by all parties, we will have disputes and differences and reports and reports without much definite result.

MR. J. R. PARK:* As Secretary of the Allied Boards of Trade and of the Joint Committee on Better Street-car Service, I was interested in hearing Mr. Morse discuss this report, which is the most comprehensive and practical that has ever been prepared on the problem of transportation in Pittsburgh.

I am not prepared to discuss the Report in detail, but I have listened with a great deal of interest to the remarks of the engineers present and I am pleased to find a sentiment in favor of doing something; in favor of getting together and putting across some of the things suggested in the Report. It presents a program upon which we can at once begin to improve transportation conditions.

I presume it is unnecessary to say that the breakdown in service last winter was one of the worst catastrophes that ever happened to Pittsburgh, damaging us to an incalculable degree in a business way, in the health and social life of the community, etc. It is possible that next winter the same thing may happen

*Secretary, Allied Boards of Trade, Pittsburgh.

again and it may be even worse. The president of the Pittsburgh Railways Company went before Council the other day and said that the increase in fares had not increased receipts. Now we know the conditions and if the increase in fares has not increased receipts, and if the equipment is not going to be improved, but allowed to deteriorate still more, conditions are going to be worse next winter than they were last winter.

I say these things speaking for the Joint Committee on Better Street-car Service, and would be pleased if you would appoint a committee of two or more men to meet with us at some time to discuss this question and see if there is not some way we can co-operate in bettering street-car conditions.

The Joint Committee on Better Street-car Service consists of forty local civic bodies, including five of the outlying boroughs; every one of the boards representing a district in the city which, of course, is vitally affected by the street-car service.

Yesterday the Committee approved finally the plan to install the express service that has been mentioned, out to Sixty-second Street, where it will connect with the two lines to Aspinwall and East Liberty, and this plan will help both the Company and the public. There are undoubtedly many more such improvements that we can support.

Mr. Lee C. Moore: I would like to ask Mr. Morse how much relief there would be if we could eliminate 60 to 70 per cent. of the automobiles from the congested parts of the city streets, referring to the proposition to change the Boulevard and run it down Duquesne Way and up Water Street and thence to Grant Street. How much would that benefit the street-car service?

Mr. E. K. Morse: That is a hard question. Out of that 25 000 representing vehicular travel there were about 12 000 pleasure automobiles and about 9000 trucks, while the remainder were horse drawn vehicles The horse drawn vehicles and the trucks are going to stay. You might get those pleasure automobiles on the wharf. That was the object of showing the wharf —17 acres, that ought to be storage ground. It is ridiculous to

park cars for from 30 minutes to 7½ hours on our business streets. If you look at the Report you will find that we recommend only two things—one is that no automobile be allowed to park during business hours on a street where there is a street-car line—and if they are not allowed there at all there is no question of their infringing—and the other is that no vehicle shall park on Oliver Avenue and Sixth Street down as far as Duquesne Way. If Oliver Avenue is to be maintained as a thoroughfare, the traffic must be kept moving and stops cannot be permitted.

Mr. P. N. Jones:* I came to-night to listen, not to talk. If I must say anything I guess I will have to discuss the discussion because Mr. Morse has been very gentle with the traction company and has given almost his whole attention to wagon and automobile congestion down town, and to the necessity of widening streets.

The discussion, however, by Mr. Park and others, I think should have some attention. The traction problem is one that I sleep with and dream about, and the efforts of Mr. Park and others for the last six or eight weeks have not made the problem any easier.

I want to correct just two little matters that have been brought up to-night. One is the statement that the average length of ride per passenger in Pittsburgh is three miles. We have a Traffic Bureau which, from many observations made in the last five years, tells us that the average length of ride per fare in Pittsburgh, taking into consideration also the zones outside the city, is 3.5 miles It might be interesting to note that the average length of ride in Cleveland on the same basis is 2.19 miles, so that the fare in Pittsburgh is actually lower than it is in Cleveland per passenger mile, not considering in the comparison the many hills and narrow streets and twists and turns in Pittsburgh that Cleveland does not have.

The other idea which I thought needed correction was one brought up by Mr. Godfrey in the discussion. He advocates

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additional loop lines with a loading station somewhere near the outer part of the loop where people would transfer from branch lines. If there is anything that a practical railroad man does not want, it is a loop line such as our present Shady Avenue line or Bloomfield line. In order to operate street-cars on time it is necessary to have some point during each trip where the cars can stand and have what is known in street railway terms as a "lay over." It is absolutely impossible to operate cars on time because of the great number of interfering vehicles, and the operation of cars on time can be approximated only if there is some point where these delays can be compensated for. In other words, at the end of the line the cars are supposed always to be started on time and to get back as nearly on schedule time as possible. In a loop line, such as the Shady Avenue route, there is no end. We have the cars lie over on the bridge near Shady and Penn Avenues, East Liberty, but there are always passengers on the cars at this point who desire to proceed. However, this point has been selected because there are fewer passengers here than elsewhere. If transfer stations are to be provided at various points, as suggested by Mr. Godfrey, routes should be run directly from the city to these loading stations and back again over the same street. Otherwise there will be people on the car at the loading station who will want to proceed with it before its "lay over" is completed and there will always be a great deal of dissatisfaction from such passengers because the car waits.

The new combination of routes on Butler Street out as far as Sixty-second Street, explained this week in the press, is the proper way to lay out a loading station—that is, one line of cars runs on Butler Street from Sixty-second and Butler to the city, and is fed by branch lines from Aspinwall and East Liberty, which enter the loading station at the side of the street near Sixty-second and Butler, and then turn back to their respective communities.

Another matter—one of the charts to night showed that the so-called "riding habit" in Pittsburgh is not as great as in other cities. I wish you engineers would just think a moment about the other cities mentioned. Pittsburgh has more subur-

ban railroad traffic than any other city with which I am familiar. You cannot get directly down town in Detroit on a steam railroad train. In Pittsburgh there are 10 or 11 different steam roads that furnish just as short a path to the down-town part of the city as the street-car lines. I suppose Mr. Morse is right in his statement to night that only 31 000 people each day use the trains. I always had an idea that there was more than that amount of steam railroad travel.

Mr. L. P. Blum: I should like to ask Mr. Jones exactly what is the theory of this transfer station at Sixty-second street.

Mr. P. N. Jones: There have been three lines of cars between Butler Street and the City of Pittsburgh—one operating from Aspinwall through to the city; a second operating from East Liberty through Morningside and along Butler Street to the city; a third operating from Sixty-second and Butler Streets to the city. It is not possible to gear all these lines together in such a way that the cars would be evenly spaced on Butler Street unless the same number of cars should happen to be operated on each of these three lines; but inasmuch as the traffic is different on the two branches, and still different on the main trunk-line, there is not the same number of cars on each of the three lines, and therefore quite frequently two cars had to be scheduled close together while a car of the third line would be possibly five or six minutes behind them. The Engineering Conference Committee, which is now studying the situation here, agreed that better service would be furnished the community if one line of cars were run on Butler Street and one line of cars on each of the branches.

It might be interesting to know that one of the difficulties with the Pittsburgh traction problem is due to the direct routing of so many car lines between every community and down-town Pittsburgh. In many cities certain residents cannot get down town without transferring—in other words they are compelled to change cars. In Pittsburgh, there is almost no community which does not have its direct car line from its own district down to the business section, which results in a great many different lines of cars on a street, and means fewer trans-

fers than in other cities and more difficulty in equally spacing the cars on the main throats. For example, in Pittsburgh there are 64 different lines of cars that come down town. In St. Louis there are only 30, and in Cleveland, I believe, 22.

The new Butler Street schedule will require 42 cars instead of 40 as previously, in the evening rush hour, and this will result in five per cent. more service in cars and, since the cars are somewhat larger, more than five per cent. net improvement.

A number of stops on the line have been cut out. A street car is geared to run about 23 miles an hour on the level. The cars in Pittsburgh make on an average only 8.9 miles an hour. The rest of the time is taken up in stops and in loading passengers. It evidently will be of very great benefit to the public to eliminate every stop that is not absolutely essential.

Mr. Edward Godfrey: The loop system I suggest would have a terminal at Point Breeze where cars could be put on their schedules the same as at Sixty-second Street.

Mr. P. N. Jones: The objection to that is that somebody would be on the car. If it went out Fifth Avenue, somebody would want to go over to the Penn Avenue district and he would object to sitting in the car and waiting.

Mr. Edward Godfrey: You could have a transfer station at Point Breeze.

Mr. P. N. Jones: That would not work satisfactorily because passengers sometimes refuse to transfer when requested to do so.

Mr. P. E. Hunter:* As one who was seriously affected financially by the street railway conditions last winter, and greatly interested in Pittsburgh generally and more particularly in the West End district—and who has studied conditions around here and also observed them in other cities—I read with appreciation Mr. Morse's report. I have never heard

*President, Independent Bridge Co., Pittsburgh.

Mr. Godfrey's arguments before but I have been arguing along the same line in many respects. I can not help thinking that a loop line in the down-town district would be a great remedy for conditions in Pittsburgh. I think people have been spoiled to a great extent by being dropped off at their very doors, but I am convinced that if a loop line were run in the down-town district along Liberty Avenue to Second Avenue, down Second Avenue to and over the Pennsylvania Railroad, thence over to Penn Avenue at some point between Eleventh and Thirteenth Streets, looping the outlying lines at points adjacent to the down-town loop, and making transfer stations at proper places; the conditions we have experienced in Pittsburgh would be eliminated to a great extent. There is no question that we who live out in the East End could make very much better time to the down-town district if the cars were not delayed through this congested triangle down town by the wagons, automobiles, etc. If these car lines were routed into a point down town such as suggested, and transfers made such as are made in the New York subway so we could practically step from one car to the other, I believe we would be able to get down town in half the time it takes us now.

Furthermore, too many stops are made. Stops ought to be made not less than three blocks apart. We all understand the wear and tear on equipment due to frequent stops, and reducing the number would benefit the operating company and it would also save the time of the passenger and the cars; and we would all be willing to walk a few blocks to save that amount of time.

Also, in regard to congestion, I believe that there should be an ordinance providing that any time a car is stopped—whether it is blocked by a wagon, automobile, truck or other means—a flag should be thrown out and the doors opened, for oftentimes that would make possible the exit of passengers who would otherwise remain on the car because they could not get out.

I do not quite understand why Mr. Morse did not consider that down-town loop or did not report on it. It seems to me if he were to put into effect the two systems he suggests

they could be easily turned around to follow a down-town loop to a point beyond Grant Street and thence out to the East End district. It also seems to me that from a political, financial, or any other standpoint, any system which is attempted in the way of putting in Pittsburgh either an elevated or a subway, which will cut across the town and eliminate the possibility of other lines coming in and connecting with it, will not meet with the approval of Council or of financiers and will thus delay operations. As I see it, the quickest means of getting relief would be from the down-town loop, which would not interfere in any way with the other proposed schemes of Mr. Morse. I heartily approve of his street widening, etc. I think it is necessary and can be accomplished, but the down-town loop is the most highly essential proposition of them all.

Mr. E. K. Morse: I am glad that question was brought up. It was touched upon and answered in the main Report and in the synopsis. The reason there was not more space given to it is the fact that if a two-track loop were built as suggested—all those things were taken up and discussed, and the figures and plans are in the office of the Transit Commission — the absolutely necessary widening of the city streets would cost not less than $5 000 000. The first period of construction from the South Side to Dallas Avenue would cost $7 050 000. A two-track loop would accommodate only 200 cars an hour while there are practically 700 cars coming in to the business section in the extreme rush hour. No subway loop to take care of the surface traffic should be designed without anticipating at least 800 cars coming in, which would require a four-track subway loop and would add between $8 000 000 and $9 000 000 more to the cost of the subway loops to handle surface cars.

If the second period, as suggested in the Report, were built from the North Side to the East End district transferring all those lines at the terminals and the main points between, as is done in every other rapid transit system in the country, it would take off 64 per cent. of the surface cars coming in to the

down-town district to-day and at the same time provide rapid transit. A loop down town will never provide rapid transit though it represents an expense equal to the total cost of periods 1 and 2, which would provide rapid transit. That is the reason we did not approve of the loop, though we designed three loops and made three estimates.

Mr. P. E. Hunter: Notwithstanding the statements you have heard I am absolutely convinced that a four-track system as proposed will take care of all this traffic we have to-day and for 30 years to come; and not only that—we are not considering exclusively the rapid transit as applied to the city of Pittsburgh, but considering that congested condition in the down-town district as affecting vehicles, accidents, etc., as well.

Mr. E. K. Morse: I should have added that the down-town loop would not pay the operating company one cent, and the city would have to pay the full amount in additional taxes.

Mr. P. E. Hunter: While it might not pay the operating company one cent; at the same time, the saving in the number of cars which would be necessary to operate on the outlying lines, due to the fact that they would not be delayed, the lighter operating cost and better service, would be sufficient for any business man or corporation.

Mr. E. K. Morse: It pretty nearly ruined the Boston system.

Mr. Samuel E. Duff: I do not think the gentlemen of this Society understand just what the Society has done in this matter. Immediately after Mr. Morse's Report was published, the Civic Affairs Committee, of which I am Chairman, held a meeting and reported to the Board of Direction a recommendation that the Society should endorse the first two sets of recommendations—street widening and the prohibition of automobile parking. Our report was approved by the Board of Direction, with the additional instruction that the report and findings of the Civic Affairs Committee be presented to the Mayor and Council. This was done; so the Society stands committed to an absolute endorsement of those two sets of recommendations.

It seems clear that all the members of this Society should follow Mr. Morse's suggestion — examine this Transit Report, study it and talk about it. It is certainly true that these two recommendations we have approved can be carried out by the municipal authorities at once. I do not know what the Society can do to expedite matters, but if we try to do something, more will be accomplished than if we stand by and do nothing.

Author's Closure: The author has listened to the discussion with interest, but will touch upon only two of the subjects mentioned, as it is getting late. Mr. Godfrey has outlined a most comprehensive elevated system, but as that part of the subject has not been presented it is only necessary to say in reply to his argument that there is not a single rapid transit system that can be laid out—long, short, elevated or subway, or both combined—that will to-day pay fixed expenses and a reasonable return on the investment. To complete as extensive an elevated system as he outlines would bankrupt any concern that ever attempted to build and operate it, and any system that is built with the full knowledge that it will not pay for its maintenance and yield a fair return within a few years should never be attempted. The present service company is a most elegant illustration of what it means to carry a large financial burden, so great that a deficit is constantly in sight.

In regard to the loop mentioned by Mr. Hunter, the Transit Commissioner did very carefully consider the question of a subway loop down town but when it is considered that a subway loop down town will cost between $9 000 000 and $10 000 000 and will not bring in any additional revenue who is going to pay for the deficit? Boston is the best example of a service company at the very edge of bankruptcy; and this notwithstanding the fact that the Boston system is financially clean, and has not been watered. Yet they have built so many subway loops and passages under the city of Boston, and then loaded them upon the traction system, that either the city or the commonwealth will have to assume the management and ownership. The effort is now being made to bring this about, and bills are now introduced in the Legislature of Massachusetts, which it is believed will be passed, creating a trusteeship.

If periods 1 and 2 calling for rapid transit recommended by the Pittsburgh Transit Commission are put into effect, 64 per cent. of the surface lines now entering the city would not enter at all. The passengers would be transferred at the terminals to rapid transit lines entering the city, and the surface lines would be turned back immediately. If even 50 per cent. of the cars now entering the city were turned back; if the streets were widened and a clear highway used as a "by-pass" around the city—all of which can be put into effect immediately—then the congestion in the down-town district will cease for 10 to 15 years to come, even if the problem is not solved for all time.

THE DESIGN OF GOVERNORS, WITH SPECIAL REFERENCE TO SMALL DIESEL ENGINES

By Arthur B. Lakey*

INTRODUCTION

The object of this paper is twofold: (1) to point out some short-cuts in the design, construction, and adjustment of certain types of centrifugal governors, and (2) to show the need of certain auxiliary apparatus to secure improved smoothness of running in the case of Diesel engines of small power, or such as are underrated, or are provided with only small fly-wheels.

In this connection, the paper dwells particularly on the case of small marine installations with clutches and reverse gears, such as are coming extensively into use for trawlers, work boats, etc.

The use of small Diesel engines, with their unequaled economy of operation, is extending rapidly into fields where the gasoline motor has hitherto been the favorite. Were it not, indeed, for the fact that most of the Diesel engine manufacturing capacity of the country is at present engaged on larger units, suitable for transatlantic service, this expansion would be even more rapid.

PART 1. NOTES ON DESIGN, CONSTRUCTION AND ADJUSTMENT OF SMALL MARINE GOVERNORS

Practically all governors for marine Diesel installations employ helical springs for loading, mounted either at right angles to the spindle, or surrounding it. Pendulum governors, formerly much used on small stationary gas-engines and to some extent on marine steam-engines (as the Aspinall governor), are practically unknown in the field under discussion, and all weight-loaded governors are necessarily avoided, on account of the comparatively large proportions which they would have to assume

*With Albert Kingsbury, Engineer, Pittsburgh.

Governor Characteristics. The best starting point for explaining the characteristics of governors under any set of conditions is afforded by the purely theoretical case of the astatic governor. It could not be realized in practice, even if it were desirable to do so, as it assumes friction to be absent. The restraining force—furnished, e. g., by springs or by weights acting through a suitable mechanism—is assumed proportional to the distance of the center of gravity of the masses from the center. The governor masses are, moreover, assumed to move parallel to themselves to and from the spindle, or to be of such shape as to produce the same effect.

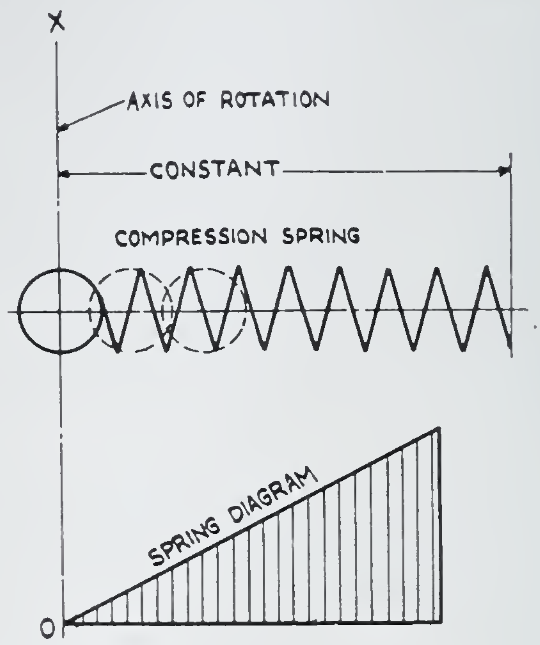

Fig. 1. Diagram of Astatic Governor.

The simplest form (Fig. 1) of the astatic governor mechanism is furnished by a mass attached to a spring of negligible mass, so mounted that when at rest the center of gravity of the mass coincides with the axis of rotation.

From the formula for centrifugal force, $F = \frac{W}{g} r \omega^2$, it is evident that F is proportional to r. But from the spring diagram, the resisting force also is proportional to r. Accordingly, assuming ω to start at zero and increase gradually, the straight-line curve through O, representing F, will move from its initial position, OA; and at length, at some definite speed ω', reach the position OB. Then, for this particular speed, the mass will be in equilibrium at any point in its travel. The slightest increase in ω will then cause the mass to go against its outer stop.

For this case (astatic governor) the spring diagram constitutes the characteristic curve of the governor, which is generally defined as the curve of *restraining force* plotted in terms of the position of the center of gravity of the mass. The definition of C-curves as above given can not be applied to ordinary shaft governors, nor to the type of governor shown in Fig. 11, without modification.

In most practical cases the restraining force, C, consists of the algebraic sum of a number of parts, as follows:

C_f = force due to springs (practically the only component requiring consideration in this paper).

C_q = force at center of gravity of governor mass due to weight of collar.

C_g = force at center of gravity of governor mass due to weight of governor masses.

It is generally desirable to plot the component curves in addition to the C-curve itself.

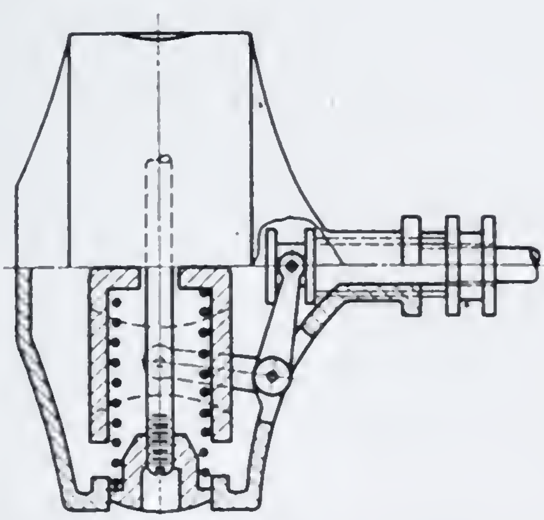

Fig. 2. Hartung Type Governor.

For a governor of the particular type shown in Fig. 2, having normal proportions, the component and resultant C-curves will appear about as shown in Fig. 3. In this figure, the center of gravity of the mass is assumed to coincide with the center-line of the pin on which it is mounted. This represents a condition which is often created purposely in this sort of governor.

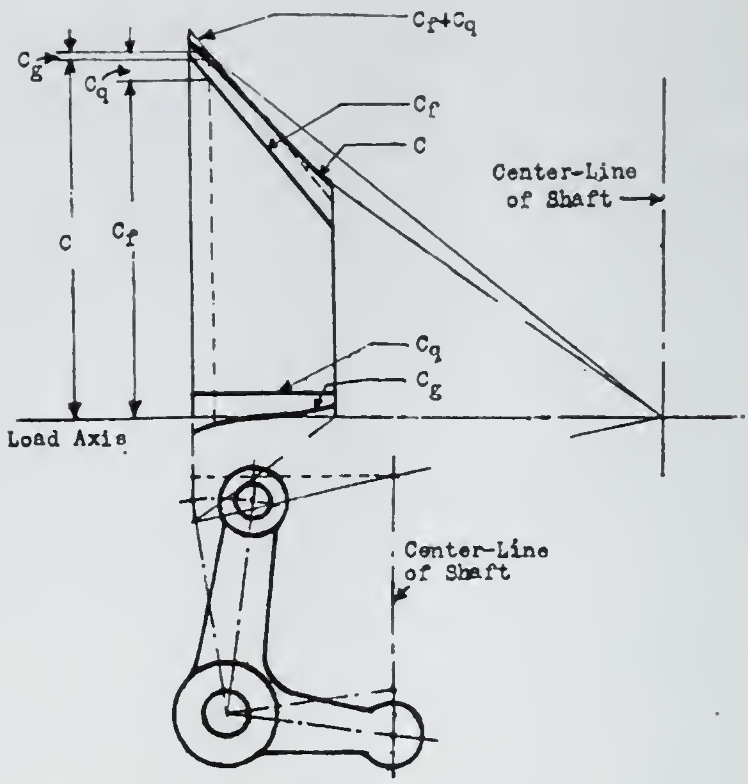

Fig. 3. Component and Resultant Characteristic Curves for Governor of Hartung Type.

For the sake of clearness, the C-curves are shown to a considerably larger scale than the lever.

The separate component curves are first shown plotted separately with reference to the zero axis, after which we have shown the graphic addition, $C_f + C_q$, and finally, $C_f + C_q + C_g = C$ (the resultant curve) which is the one from which the coefficient of speed variation is directly obtainable, as hereinafter explained. The additions are understood to be algebraic.

The use of characteristic curves and the notation here employed are according to Tolle.*

In general, the C-curve and its components are curved lines, although in many important cases the C- and C_f-curves are straight. This latter condition exists on the most generally used type of precision governor. (See Fig. 2.) For ease of classification, we shall first assume the C-curves to be straight lines, and later extend the method to include more general cases. Fig.

*Max Tolle. "Die Regelung der Kraftmaschinen," Ed. 2, 1909, Chapter 6. Springer. Berlin.

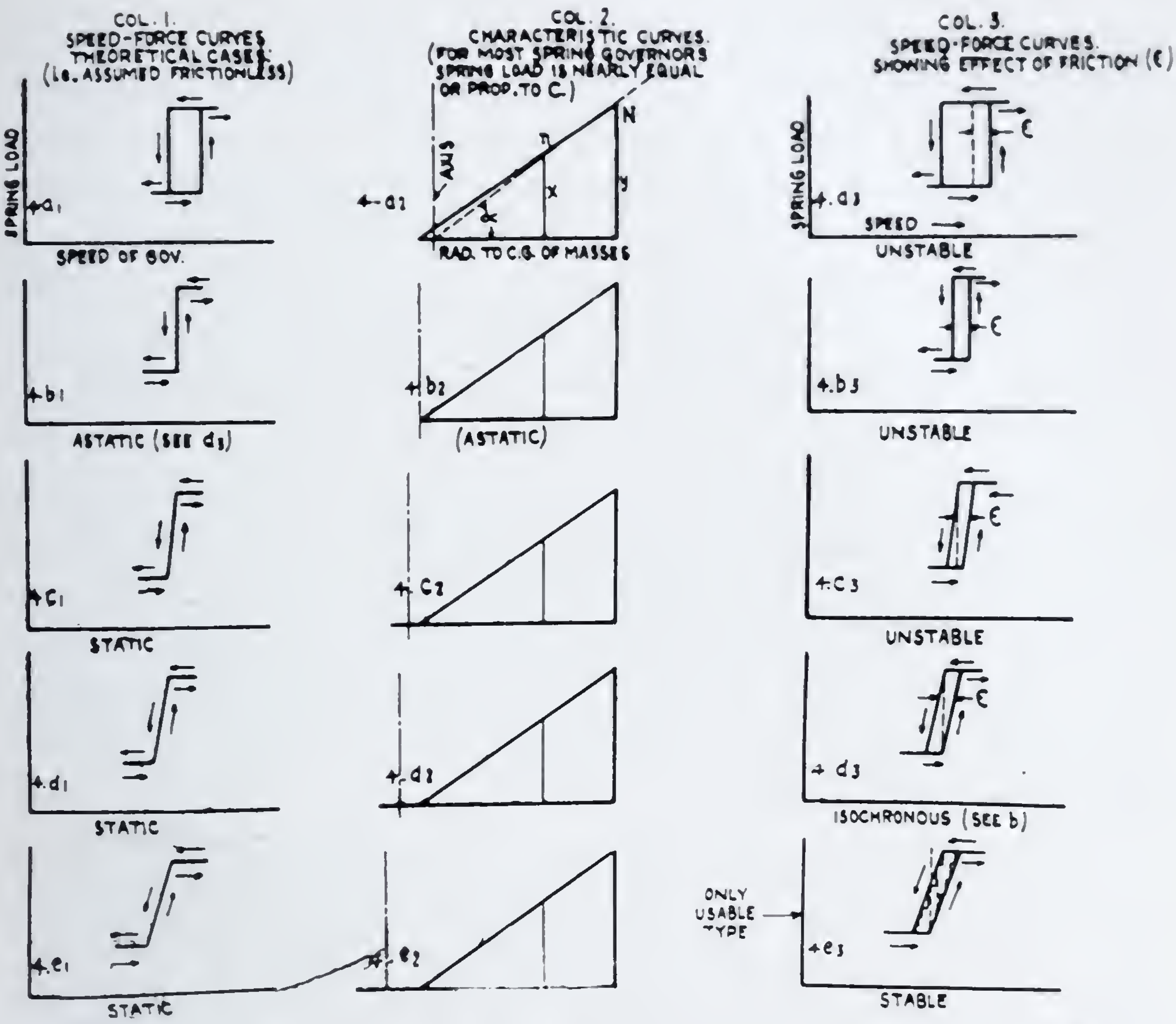

Fig. 4. Curves for a Hartung Type Governor, Showing Effect of Shifting Axis of Rotation.

4 shows, in column 2, a progressive series of characteristic curves. The "astatic" condition just described, again shown in Fig. 4-b_2, represents the theoretical boundary between usable and useless arrangements.

Fig. 4-a_2 (unstable governor) represents the case where the axis is reached before the restraining force, C, reaches zero. Our discussion of the astatic arrangement made it clear that the angle a (or more strictly tan a) increased as ω, but it is evident that a ray through the origin would in this case intersect the C-curve at high loads for low velocities, and lower loads for higher velocities.

The significance of this diagram, as shown in the behavior of the mechanism under these conditions, is as follows:

As the speed increases gradually from zero, the mass will remain against its inner stop until the limiting speed, N, corresponding to the inner position is reached. This, as will be noted,

is the greater of the two speeds for which the rays through the origin have been drawn. As soon as this speed is reached, the mass will fly out against its outer stop, with an increasing acceleration, as the net moving force will increase during its movement. (The centrifugal force increases directly as the distance to the axis of rotation, but, as shown by the C-curves, the restraining force does not increase so fast. Therefore the net moving force, which is the difference of the two, will increase as stated.)

Decreasing the speed from some still higher value, an analogous line of reasoning will show that the mass will abruptly move in from its outer to its inner stop when the lower of the two critical speeds, n, is reached. This action constitutes a bad case of hunting. It is evident that the effect of this unstable arrangement is to prevent the machine from running smoothly between the speeds n and N.

The actions above described could be represented by a speed-force diagram like Fig. 4-a_2 in an actual machine. If no friction were assumed, the vertical lines in the figure would be closer together, as in Fig. 4-a_1. The effect of friction is shown by the difference, ϵ, in the speed variation for the two cases.

The practical consideration of stability, and hence usability—unlike the case just described—demands that each position of the mass, while the speed changes continuously, shall coincide with some definite value of the r.p.m.

The diagrams (Fig. 4-c_1 and 4-c_3) make it evident that under certain circumstances the introduction of friction may change a given governor from stable to unstable.

Fig. 4 shows in the second column a series of five characteristic curves beginning with the unstable condition and ending at the stable condition. The diagram assumed is a straight line in each case, which is approximately true for the most generally used type of spring-governor, based on the original Hartung type, in which the masses move in a line, at right angles to the spindle, coinciding with the axis of the springs. It would be but slightly more difficult to make the demonstration using curved characteristics.

In Fig. 4 the spring curve and loads are kept constant and the effect of shifting the axis of rotation with respect to the stops is examined.

The angle, between the horizontal axis and the ray passing through the origin, has been shown in connection with Fig. 1 to increase with the velocity. It is readily seen that in Fig. 4-a_2 such a ray will intersect the characteristic curve at Y as w is gradually increased before reaching X. The end position of the mass therefore corresponds to a lower speed than the inner position and, as previously explained, hunting is certain to take place.

Column 1, Fig. 4, shows what results might be expected from the test of a governor, provided friction could be eliminated; while column 3 gives similar information typical of governors actually constructed. In every case as the speed increases from zero it reaches a value at which the masses start to move outward. After the outer position is reached, the speed is permitted to increase somewhat further, after which it is again reduced gradually and the point found at which the mass starts to move in. We finally obtain a diagrammatic record of the behavior of the governor, somewhat analogous to a hysteresis curve. (See diagrams in columns 1 and 3.) The manner in which the speeds are made to change is shown by the arrows.

In the "a" diagrams, the governor is unstable in both cases, although it is evident that the effect of friction is to increase the instability. The number of revolutions by which friction increases the speed difference is shown by ϵ in Fig. 4-a_3.

The "b" diagrams show the astatic condition, in which if the friction could be removed the masses would be in equilibrium in any position at some particular speed. Fig. 4-b_3, however, shows that owing to the effect of the friction the governor is still unstable.

The "c" diagrams show the case of slightly static governor where the effect of friction is sufficient to make the actual mechanism unstable.

The "d" diagrams show a slightly increased degree of stability in which with the actual mechanism the speeds of starting out and starting in are the same*—that is, the effect of the friction is just enough to neutralize the stability.

*This is the definition of the "isochronous" condition.

Concerning Fig. 4-b_1 and 4-d_3, it is to be noted that the astatic condition is a special case of the isochronous condition for which the friction equals zero.

Any further increase in the stability, as indicated by the "e" diagrams, will render the governor usable, as the effect of friction is no longer sufficient to overcome the stability.

Fig. 4-e_3 corresponds to the only usable governor arrangement. In Fig. 4-c_3, 4-d_3, and 4-e_3, we have each transition curve shown as a smooth line but this is not strictly correct, as the effect of the static friction of the parts is bound to make them appear as a series of irregular steps, somewhat as indicated in Fig. 4-e_3. The very fact that the static friction is subject to such capricious changes makes it necessary to allow a safe margin of stability.

In all the old types of governors the characteristic diagrams formed curves which in some cases had points of inflection. In the design of such governors it was possible, by studying all the C-curves, to have a range of choice in the value of δ, trying the various values for the throw of the masses, and endeavoring to utilize the portions of the C-curve which would give the desired results.

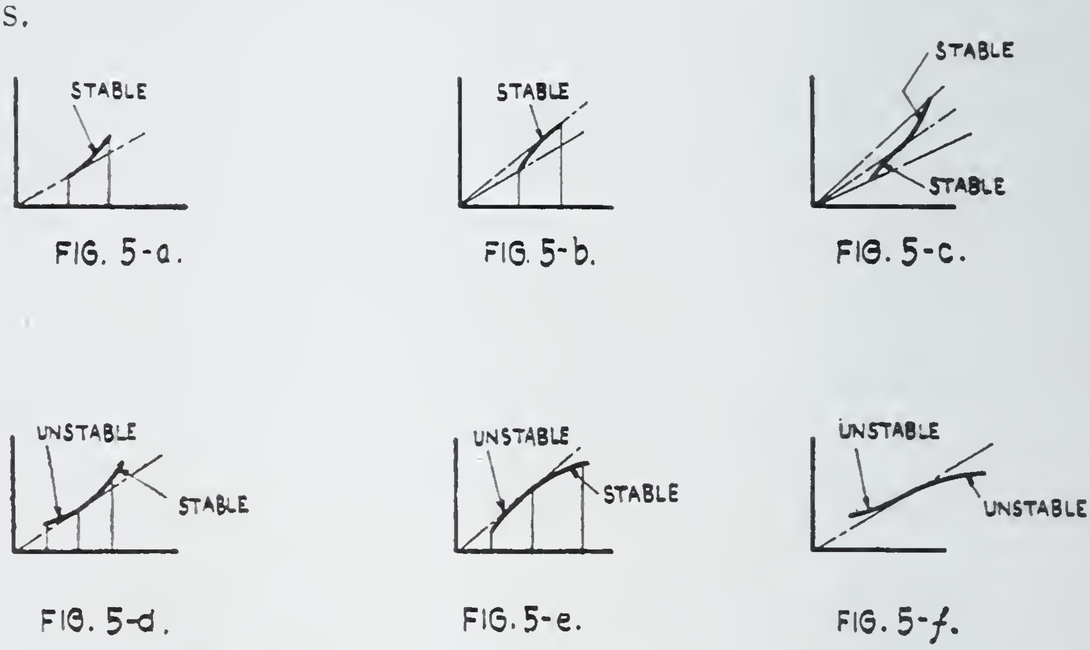

Fig. 5. Types of Characteristic Curves.

Fig. 5 shows a number of curves which might be encountered. In the examination of these curves it will be seen that the value of δ varies greatly for different portions and that it is possible, as in Fig. 5-c, for the middle portion of the curve to be astatic while the curve as a whole may be stable.

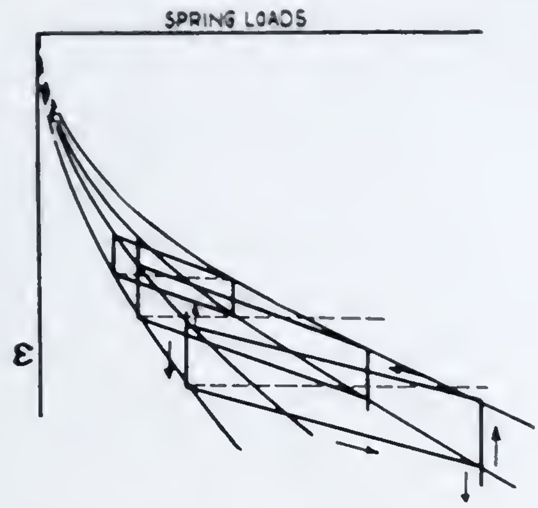

Fig. 6-a. Diagram Illustrating Method for Choosing Proper Spring for Any Desired Speed (Governor Mechanism Assumed Completely Known).

Adjustment of Governors. Fig. 6-a shows a series of speed-load diagrams similar to those in column 3, Fig. 4, such as may be obtained by testing any given governor with different springs and at various spring tensions. The corresponding corners of the parallelograms obtained by test will be found to lie on a system of approximate parabolas. For the case of zero friction, the parallelograms would shrink to a system of straight lines having their extremities on two parabolas passing through the origin. Having such a set of curves for a given governor it is easy to plot additional parallelograms from which the maximum speed variation and spring loads may be predicted directly.

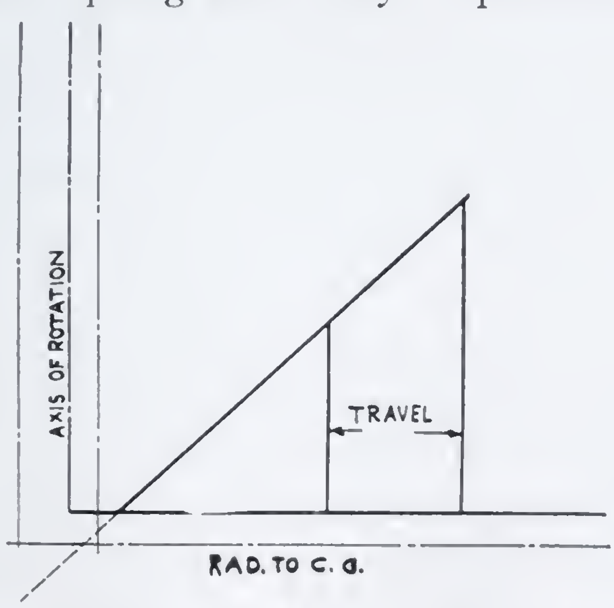

Fig. 6-b. C-Curve, Illustrating Method of Obtaining Desired Speed and δ With Assumed Spring.

Fig. 6-b shows how the governor may be given any desired characteristic. When the spring of given scale has been assumed,

it shows that by shifting the axis of rotation to the left (or right), δ will be increased (or decreased) as desired. It also shows that a similar effect can be produced by tightening or loosening the spring. Loosening has the effect of increasing δ, and tightening of decreasing it. It is also evident that δ may be altered as desired by changing the throw of the masses. (See also Fig. 6-c.)

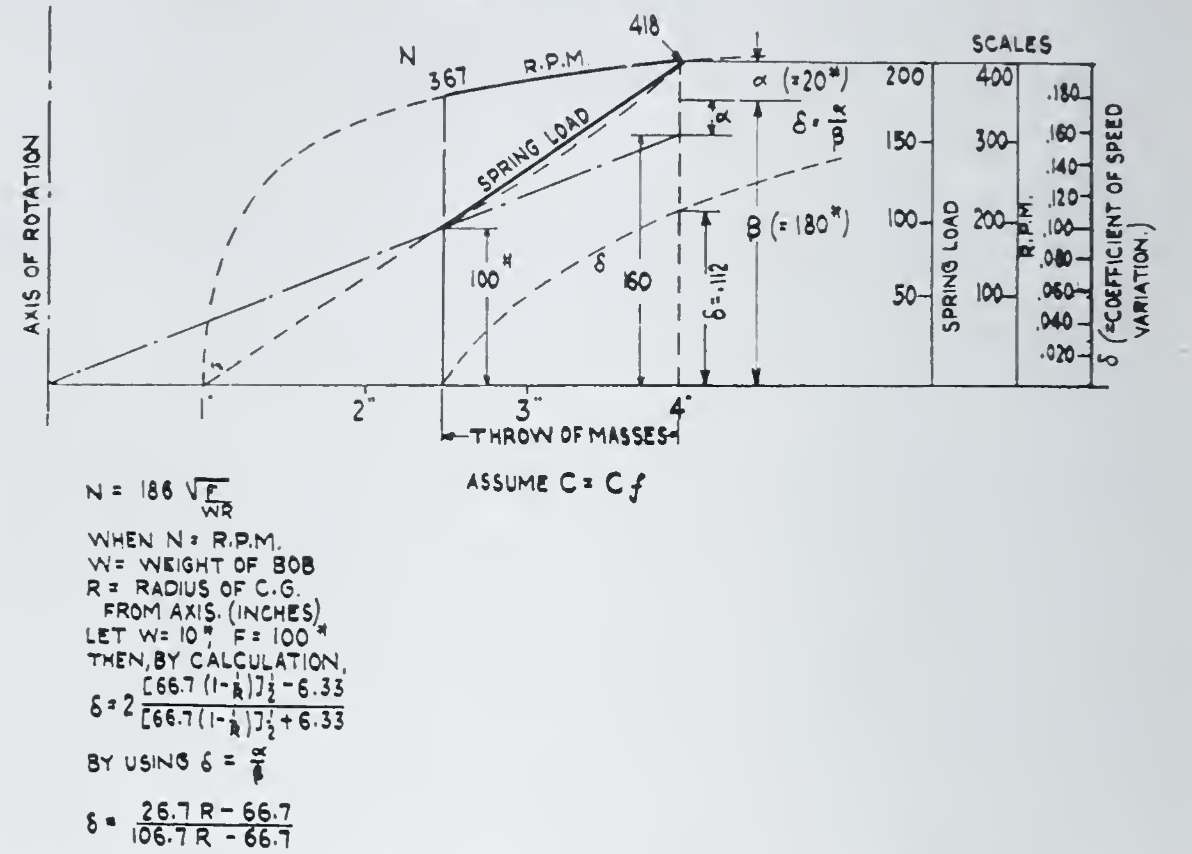

Fig. 6-c. Determination of δ from C-Curve.

Determination of C-Curve. Fig. 6-c shows some numerical results for an assumed characteristic curve. The center of gravity of the mass is assumed to start from its inner position at the radius of 2½ inches from the axis of rotation, and to have its outer stop at the radius of four inches. The curve for δ shows how the speed variation would be decreased by reducing the throw of the masses, keeping the location of the inner stop unchanged. The spring loads, which are here assumed to constitute the entire restraining force, are 100 pounds for the inner position and 200 pounds for the outer. This makes the load curve intersect the horizontal axis at one inch radius. The speeds corresponding to the inner and outer positions are found to be 367 and 418 r.p.m. respectively.

The value of δ is most easily obtained by geometrical methods. The ray passing through the origin and touching the ex-

tremity of the 100-pound line will cut the 200-pound line, intercepting upon it the distance $2a$ between itself and the spring curve. The value of δ is obtained by dividing a by β where $\beta = 200$ pounds $- 2a$.

Two formulas are given in the figure for δ. The simpler one represents the geometrical approximation, while the more complicated one is derived from the actual maximum and minimum speeds. The approximation is very close for all practical cases.

Running Adjustment of Governors. We have seen that with a given C-curve it is possible to get any desired degree of speed variation by the proper choice of the co-ordinate axes. While it is generally impracticable to alter the location of the axis of rotation in an actually constructed governor, it is an easy matter as a rule to raise or lower the horizontal axis. This affords the simplest means of finding how to adjust an ordinary governor, say of the Hartung type, to give the best regulation at the rated speed of the unit. By trying various adjustments of the spring tension, some speed will be found at which the unit will run more satisfactorily than at any other. If this speed is too far from the desired speed it will be necessary to change the scale of the spring. Supposing that the engine is found to run most favorably at 320 r.p.m. with initial and final spring loads of 200 and 260 pounds respectively. Suppose also that the proper speed of the unit is 350 r.p.m. The correct spring loads to produce this desired speed are immediately found by multiplying the old spring loads by the square of the speed ratio. This immediately gives us, for the correct spring loads, 239 pounds and 311 pounds respectively.

The simplest way in which to reduce the number of effective coils of the spring so as to give it the scale necessary for these loads is to heat it and close up some of the end coils. Another way which has often proved convenient is to make a plug, fitted to the helix of the spring, which may be screwed into one end of the latter, altering the number of effective coils until by trial the spring will show exactly the scale desired.

Both of the above methods have been shown to be greatly superior to endeavoring to make a new spring so as to have ex-

actly the desired scale.

In the foregoing explanation we have assumed that the spring is the only agency furnishing the restraining force; in other words, that $C = C_f$. In many cases, however, the weight of some of the mechanism acts in such a way as to aid or oppose the springs and this effect must be included in the restraining force.

It is very desirable to have a testing apparatus for the governors by which they may be run steadily at any desired speed before being mounted on the engine. Such a test rig is most conveniently driven by a shunt motor with an auxiliary rheostat by which the steps of the regular rheostat may be bridged.

Another method for altering the effective scale of the governor spring consists in adding an outside auxiliary spring. Such a spring may be so used as either to oppose or to assist the inside spring; but a little consideration will show that the effect of any added spring will be to increase the scale of the combination and in no case to decrease it.

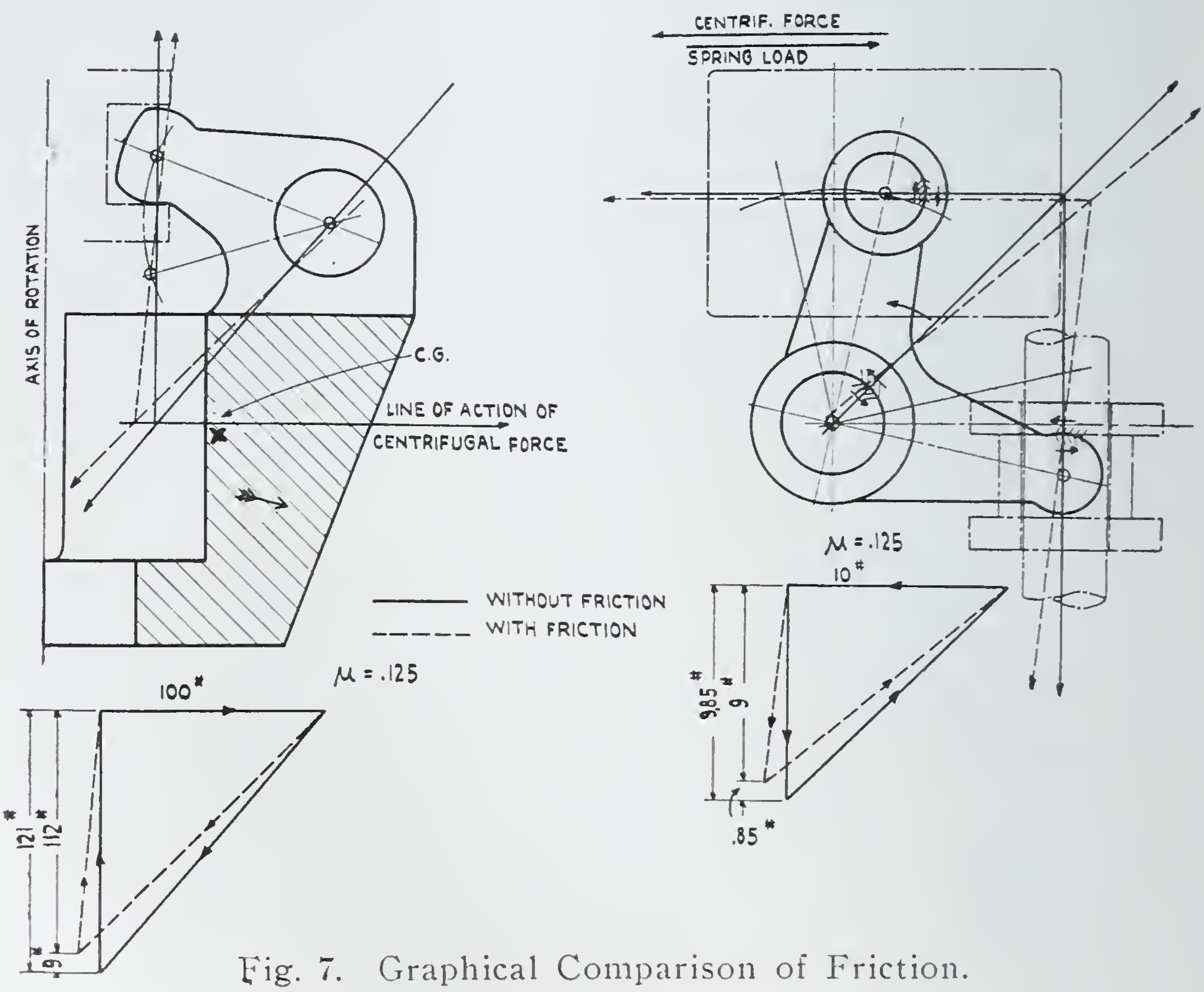

Fig. 7. Graphical Comparison of Friction.

Fig. 7 gives a graphical determination of the friction for two types of governors—first, the ordinary marine type of limit governor with conical masses; and second, the Hartung type of governor. The principal advantage of the latter type in securing low friction, lies in the fact that the mechanism in that case transmits only the force equal to the difference between the centrifugal force and the spring load, while in the former type the force due to the centrifugal action must be transmitted through the bearings before reaching the spring.

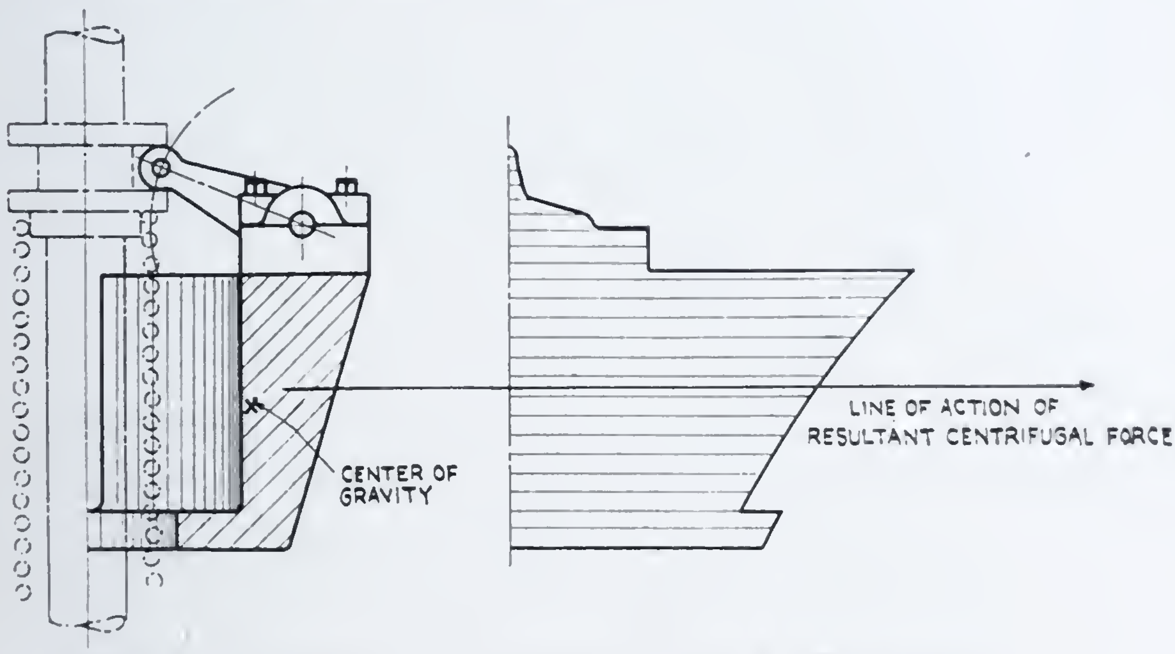

Fig. 8. Diagram of Centrifugal Force (Inner Position).

In the governor mass of the type shown in Fig. 8, frequently used on limit governors for marine work, the calculation of the force action of the mass is very complicated. Designers frequently attempt to figure the moment of the centrifugal force about the supporting pin by assuming that it acts through the center of gravity of the mass, but this is not generally the case. While the magnitude of the centrifugal force is correctly obtained in this way, its line of action will be found to be in error, the total centrifugal force being the resultant of the centrifugal forces of the small elements of the mass. As shown in Fig. 8, the so-called "diagram of centrifugal force" is obtained for any particular position of the mass by finding the center of gravity of each horizontal section and plotting the centrifugal force per unit of height. The area of this diagram will therefore represent the total centrifugal force, and the line of action may be found by cutting a templet to the shape of the

shaded portion and balancing it. For a mass similar to that shown, the resultant will be found to lie above the center of gravity for the inner position and below it for the outer position.

On account of these conditions it is very difficult to obtain the characteristic curve for a governor of this type, and the speeds may be more easily found by calculation.

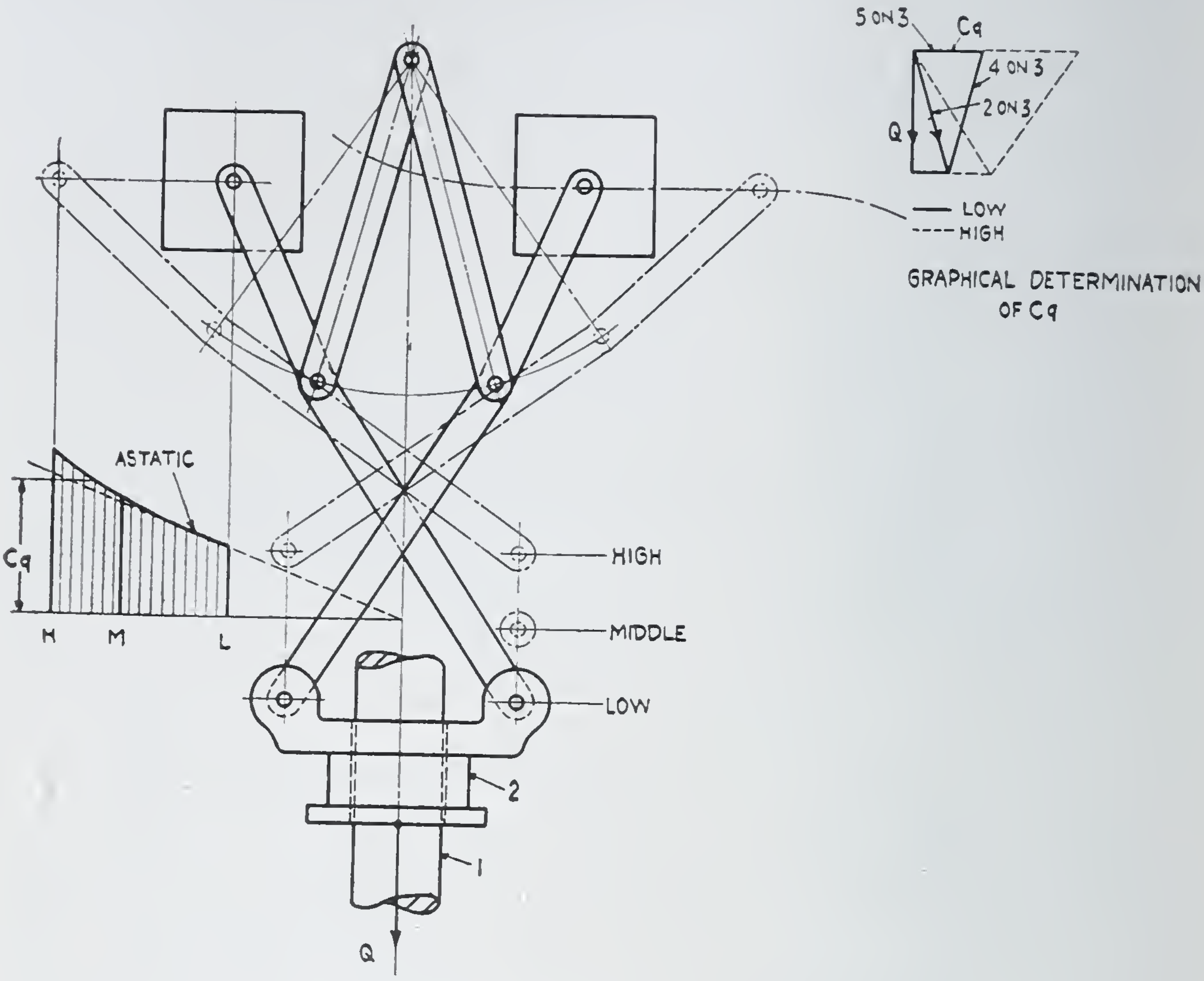

Fig. 9. Steinle-Hartung Governor.

Steinle-Hartung Governor. This is a type of governor intended for large stationary units. (See Fig. 9.) It is unnecessarily delicate for marine use. It can be shown that for any particular size of fly-wheel with a given engine there is one value of δ which will produce better results than any other. Very close regulation, such as is required on large electric lighting units, is possible only when large fly-wheels are used. Accordingly, it is useless to employ an expensive governor on work where the fly-wheel size is restricted, as the cheaper one will give just as good results with the large value of δ required under the circumstances.

In the governor under discussion, friction has been reduced to a very low quantity by the use of a straight-line motion, to cause the masses to move at right angles to the shaft, in place of the guides used on certain other types. The designers have also been successful in making the C_q-curve approximately astatic. The method of plotting this curve for a given assumed value of Q is shown in Fig. 9.

The great value of the astatic C_q-curve is that the speed of the engine may be altered within wide limits by simply adjusting the weight attached to the collar, or by using a soft spring with a large number of coils to press against the collar. It is evident that the algebraic sum of an astatic curve and a stable curve will itself be stable.

PART 2. APPLICATION TO SMALL MARINE DIESEL INSTALLATIONS

The principal difficulty met with in the control of small Diesel engines arises from the extremely small volume of fuel-oil that is pumped to the cylinder for each working stroke, especially at light loads. This quantity may often be only a few drops, and there are many conditions to be expected which will make this supply very erratic.

The qualities wanted in the power-plant of a trawling vessel as regards handling are easily met by gasoline engines; but the governors on such motors have only a light butterfly valve to actuate, while their counterpart on the Diesel engine must move a part of the mechanism of the fuel-oil pump, which, on account of the high working pressure—about 1100 pounds to the square inch—must unavoidably be of rather heavy construction and have closely fitted parts to avoid excessive leaking of the fuel.

The normal operation of these trawlers while on the fishing grounds requires drifting for spaces of something like half an hour, alternating with short periods of slow ahead running. As the engineer is customarily required under such circumstances to help in handling the nets, the engine must be capable of running idle without excessive hunting for the greater part of the time, but be able to take the load without touching the "throttle" (more properly "fuel control") when the clutch is engaged.

It is obvious that if a Diesel engine equipped with a constant-speed governor were tried in this service, it would be necessary to throttle it down after disengaging the clutch and to speed it up before re-engagement. It is found in typical installations—for reasons which we shall give—that the engine will not run smoothly without load, especially at low speeds, unless some means is provided aside from mere "throttling." In addition, the elimination of this hand control enables the operator to give more attention to other duties. By fitting a suitable adjustable-speed governor, however, these difficulties are overcome. The speed of the engine can be controlled absolutely by setting the governor-adjusting lever or screw, as the revolutions will remain practically the same while the engine is running light as while it is driving the ship. This eliminates the hand fuel control. When trawling under such conditions, the operator has only the clutch to handle, and the engine will run smoothly at any desired speed, down to say half speed, whether driving or running free. The only precaution necessary is that of engaging the clutch gradually, so as not to stall the engine. It will be noted that such an adjustable-speed governor presents two problems:

1. The provision of means to enable the engine to run smoothly at light loads and low speeds.

2. A suitable arrangement of adjustable springs to constrain the engine to run at the desired speed, whatever the load, up to the maximum corresponding to this speed.

Difficulties Connected with the Governing of Small Diesel Engines. Assume a four-cylinder engine, 100 horsepower—25 horsepower per cylinder—350 r.p.m., using oil of 18 000 B.t.u. per pound, 35 per cent. thermal efficiency.

With 25 horsepower per cylinder and 174 working strokes per minute, the work per cylinder cycle $= \dfrac{25 \times 33\,000}{175} = 4720$ foot-pounds; $=$ 6.06 B.t.u. Therefore, the oil per working stroke $= \dfrac{6.06}{18\,000} = 0.000337$ pounds.

One cubic inch of oil (density 0.87) weighs 0.0314 pounds, and the volume of oil per cycle per cylinder $= \dfrac{0.000337}{0.0314} = 0.01072$

cubic inch. So, assuming 250 drops per cubic inch, the number of drops per cycle per cylinder = 250 × 0.01072 = 2.68 drops.

The corresponding figure for a 450-horsepower engine at 350 r.p.m. would be 12 drops per cycle per cylinder. Diesel engines are usually designed for maximum pressures of 500 to 550 pounds per square inch. In practice it is found that such engines will not fire with any degree of regularity if all cylinders are throttled alike, at less than about half speed under ship conditions, even with fairly large units—up to 150 horsepower per cylinder, and probably larger. It is bad practice to use a higher maximum pressure than cited—500 to 550 pounds per square inch. If a lower one is used the difficulties become even more marked at the low speeds, as the limit of reliable running is reached sooner as the speed is decreased.

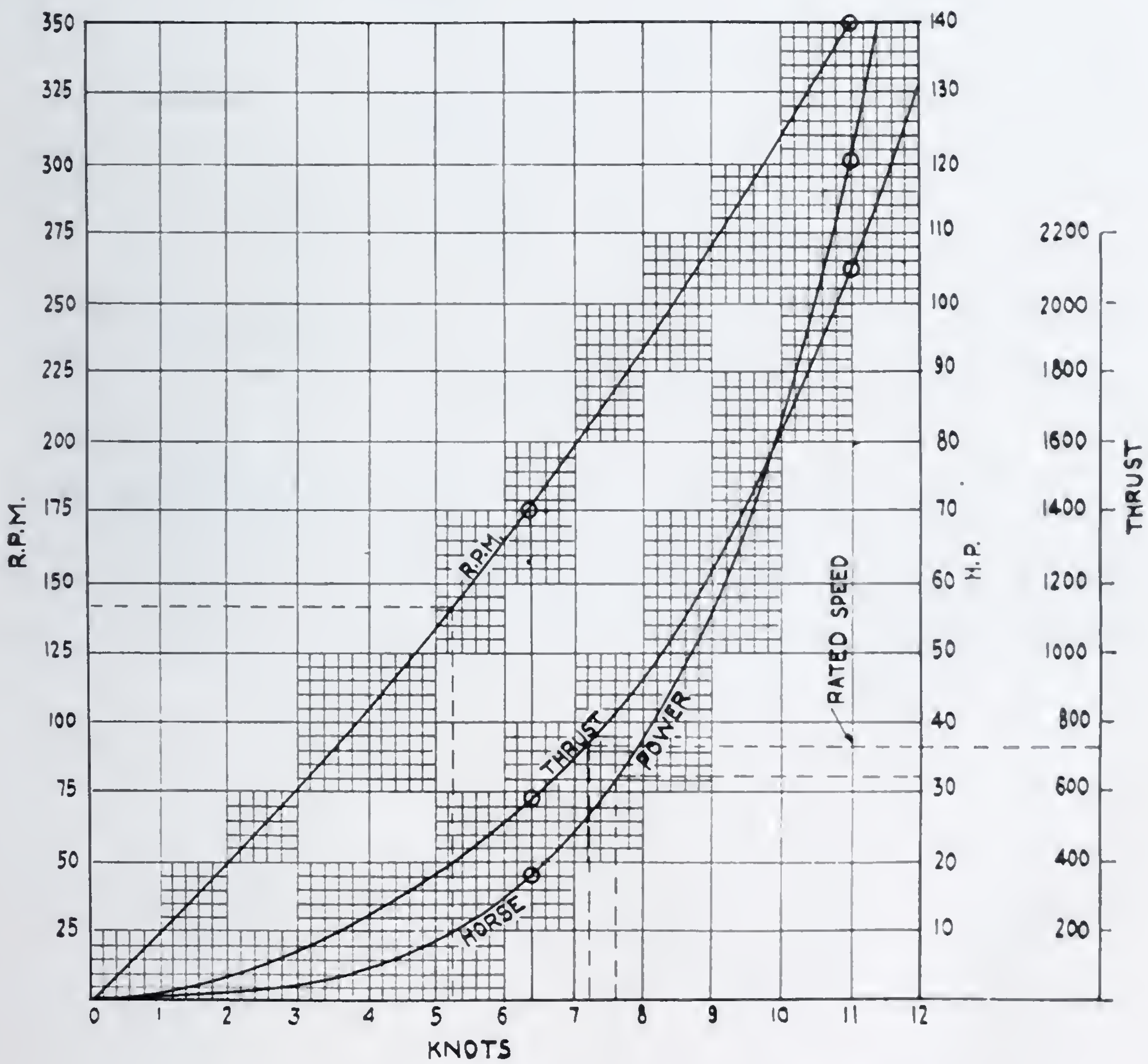

Fig. 10. Speed-Power Curves of Typical Work Boat About 75 Feet Long.

In Fig. 10 are given the speed-power curves of a typical work boat about 75 feet in length, equipped with a 120-horsepower motor. The revolutions of the propeller come somewhere near being proportional to the speed of the vessel, due to the fact that the propeller efficiency does not vary greatly through the ordinary range of speeds. The horsepower curve shows that, for the case assumed in connection with the adjustable-speed governor for such an installation, the horsepower at half speed is less than one-sixth of the rated horsepower. In such a case, the indicated horsepower might be around two-fifths of the brake horsepower at full speed, resulting in a roughly proportionate reduction in the fuel supply—a very small quantity even at full load.

Two of the principal causes for erratic variations in the fuel supply on small constant-pressure engines are air pockets in the pump, pipe, and spray valves; and elasticity of the pipe lines. Since the discharge pressure is generally around 1100 pounds per square inch, a very small air pocket can cause considerable trouble, as during the suction stroke the minute quantity of pocketed air will expand to many times its previous volume and may easily amount to more than the entire supply of oil which should be retained in the pump chamber.

Adjustable-Speed Governors. In the Steinle-Hartung governor, elsewhere described, as well as in the Tolle type, it is possible to secure adjustment for a wide range of speeds without sensibly changing the value of δ.

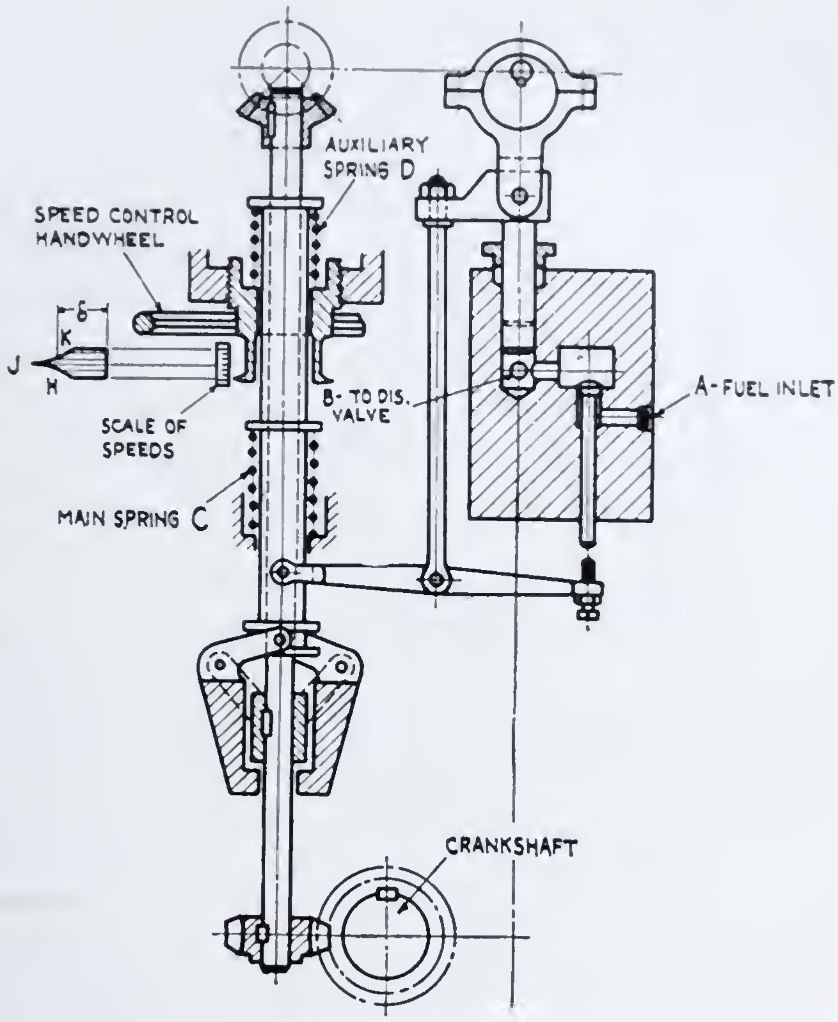

Fig. 11. Governor with Conical Masses.

In the smaller type of governor, with the conical masses, such as is frequently used for marine work (See Fig. 11), the value of δ cannot be kept constant when the auxiliary spring (as D, in Fig. 11) is used, but it is possible to make the initial and final values (H and K) of δ take a suitable value. In this case, however, the intermediate values will necessarily rise to a considerably greater magnitude. This is illustrated by the C-curves shown in Fig. 11-a.

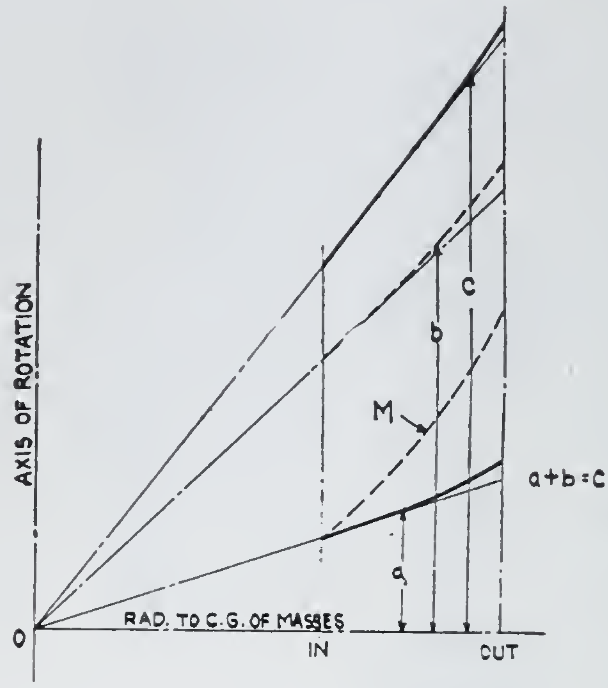

Fig. 11-a. Characteristic Curves (Nearly the Same as Spring Curves) for Fig. 11.

a. C-Curve, Main Spring Only (Engine Runs at Low Speed). See "H," Fig. 11.

b. C-Curve, Which Auxiliary Spring Would Produce if Acting Alone.

c. C-Curve Produced by Both Springs When Fully Active (Engine Runs at Full Speed). See "K," Fig. 11.

Having chosen suitable values for the combined spring loads to produce full speed, it is clear that the corresponding loads for half engine speed will have one-fourth the value of the original figures. Since, in the latter condition, the inside spring alone is active, it will be designed to give these smaller loads; and the auxiliary spring, which gets fully into action when full speed is reached, will have loads three times as great, so that their combined tensions will give the proper effect.

For intermediate positions the setting which will produce the greatest value of δ will occur when the outside spring begins to take hold at the instant when the governor starts out. In this case the speed corresponding to the end position still has its least value, while that corresponding to the outer position has already greatly increased. This condition is shown by the curve corresponding to the maximum value (J) of δ represented.

Owing to the erratic firing of the cylinders at light loads, trouble will inevitably be encountered if an attempt be made to run the engine with this arrangement in the absence of some auxiliary device. The fuel pump shown in the diagram is for

one cylinder only, but actually we will have four or six such pumps with a corresponding complication in the tappet mechanism by which the governor controls the quantity of fuel pumped to the cylinder.

The obvious method to be followed in order to make the engine run smoothly at light loads is to make the governor shut off entirely the fuel supply to some of the cylinders, taking care that those cylinders which are left in operation will give a good distribution of turning moment. The cylinders which are left in operation will work at a much higher fraction of their normal load than if the entire engine were firing. It is easy to understand that they will be very much less sensitive to erratic changes in the oil supply under these conditions, and that they will all fire alike giving tolerably smooth operation. It is better to have half the cylinders working properly than to have all of them working irregularly.

DISCUSSION

CHAIRMAN: Opening a periodical at random this evening I found a letter from a man who was in trouble and asking for assistance in running Diesel engines. It appeared that his engine was afflicted with what he called "butts." I do not know whether or not that is a commercial term. It is new to me, but I dare say the author will recognize it. Throwing off the load suddenly, the engine would race for five or ten minutes before settling down; so it is evident that these points the author has brought out are not pure fiction. They seriously affect the service of the engines.

We shall be glad to hear any discussion of the paper. Mr. Egan, from your wide experience you must have had something to do with Diesel engines.

MR. FRANK L. EGAN:* I have had some Diesel engine troubles. I have had some experience in governing Diesel engines for permanent speeds, but in marine practice I have had no experience at all. I think Diesel engines in this country date from the time Busch-Sulzer introduced them in St. Louis and put two in the Citizens' Traction Terminal Building in Indianapolis, Ind., and two in Ft. Wayne, Ind., and; I think about eight up in Wisconsin. Part of my difficulties were governor troubles due chiefly to the pumps handling the oil. We had a great deal of trouble with the oil pump plungers, which would pit and score. These were constant-speed units, in most cases direct connected with the line-shaft; also direct connected with two clutches.

It is some time since I have had anything to do with the question of characteristic curves. It dates back to some years ago when they were using the Ball-Rites coalition type belt and with that type of belting we were able to run at constant speeds and our trouble was that it was rather difficult to get a sufficient variation between a no-load speed and a full-load speed. We

*Engineer on River Equipment, Carnegie Steel Co., Pittsburgh.

had more trouble in getting enough speed variation to handle alternating current generators than we did with the governor. The outcome of the matter was that several hundred governors of the Franklin-Ball type were installed, and they are still running and are very successful.

To avoid friction John B. Sweet designed the Sweet governor. It was developed in Canada and by the time it was introduced into the United States I became connected with it actively for 4½ years and at that time we were easily able to get a governor that would run flat from no load to full load. There were two of them in the Oliver Power Building, two in the National Tube Building on Seventh Avenue and two in McKeesport. The McKeesport engines ran flat from the time they were installed, and continue to do so. The governor is a flat leaf-spring with a weight connected directly to the spring and with two bearing joints. When it was first brought out we used an oil dash-pot to limit its fluctuation but we soon found out that this was not necessary. It has a weight, as part of the eccentric, that comes in after the governor has gone to the extreme position of its range.

It was running through my mind, while considering the curves, that we had in the Armstrong-Sweet governor what has been described as an ideal condition. A weight on the end of a flexible spring and the centrifugal forces and the spring forces not transmitted through any bearings whatsoever. This governor was finally brought to a point where it was manufactured in the shop, where we would screw the spring down to a certain point and send the governor out, and we could hit it within four or five revolutions on a 200-revolution engine. And with the aid of a direction sheet we could explain it to the operating engineer so that he could get the regulation down to 1 or 1½ per cent. which at that time I considered something phenomenal, and which is equal to the best you can do to-day with any type of governor. When we had a great deal of trouble with the electric end of a parallel-connected alternator system, we used to fit the shaft governor with an electric motor supplied with current by collector rings so we could slightly increase or decrease the speed after the alternators were in.

Fig. 11 shows two weights mounted on a vertical shaft and two springs joined above. You transmit your centrifugal forces through pins and connections to the collar. Does not that make a certain amount of trouble which might be avoided by means of a direct connection as in the Hartung governor? Would it not be possible to have a direct connection and have your adjusting spring superimposed on that?

Mr. Arthur B. Lakey: The Fairbanks-Morse governors employ masses similar to Fig. 11, with springs arranged to tie them together directly.

Mr. Frank L. Egan: I am interested in this spring proposition. Is not the usual practice to handle these engines with this speed-control governor without a throttle?

Mr. Arthur B. Lakey: It is something that has not been introduced very largely but it is working very well in several installations of which I know. There is an extension connection for the governor out to the deck worked by a lever, and also a handle for engaging and disengaging the clutch. The engineer can go out and help handle the nets, letting the engine run free at about half speed for as much as half an hour. When the work with the nets requires it they give the boat a little shove ahead. To do this it is merely necessary to engage the clutch gradually. The engine will keep running at substantially the same speed as while drifting.

Mr. Frank L. Egan: I was considering a boat such as a fishing boat or a tugboat, controlled from the pilot house; and that control should be a speed control and clutch, and the steering equipment should also be in the pilot house. That is what I was interested in, primarily.

Mr. Arthur B. Lakey: The same thing is used in harbor boats and boats that have to run at almost any speed up to full, including running free a large part of the time.

Mr. Frank L. Egan: Take a tug with a barge; does the engineer have to speed the motor up to any extent to keep it from stopping when he engages the clutch?

Mr. Arthur B. Lakey: No. He can enter it very gradually and the motor picks up its load. As soon as the clutch starts to engage, the speed falls off slightly causing the governor to turn on the pump, after which the engine quickly picks up.

Mr. Frank L. Egan: Does this type of governor apply to the reversing type of motor?

Mr. Arthur B. Lakey: To the best of my knowledge, it has been applied in marine work only to the type with reverse clutch, but I see no reason why it could not be applied to a reversing motor.

Mr. Frank L. Egan: Reversing directly would not affect the governor?

Mr. Arthur B. Lakey: No. For small units, as low as 240 or 180 horsepower it seems to be preferable to use a clutch and gears. Regarding larger units, however, I have seen engines up to and above 700 horsepower that reverse very successfully.

Mr. Frank L. Egan: Do the reverse gears pick up readily for astern running?

Mr. Arthur B. Lakey: The gears are generally designed for a reduction of speed on the reverse. It is therefore impossible to get a very high power on the reverse, and they protect themselves in that way.

Mr. Frank L. Egan: I would like to get practically as high an efficiency on the reverse as when going ahead. Our backing power is just as important as going ahead, or even more so.

Mr. Arthur B. Lakey: That is new to me. We have never had a case in my experience where we cared about much power when backing.

Mr. Frank L. Egan: Where you are handling barges and get in tight places the reverse is just as important, and sometimes more so.

CHAIRMAN: It would be interesting to know just how these small quantities of oil are introduced into the cylinder and how handled after getting by the governor. I dare say some of the members are not familiar with those matters.

MR. ARTHUR B. LAKEY: The fuel line from the pump to the cylinder is generally somewhere around 3/16- or 5/32-inch copper or steel tube and of course it takes quite a long time for a charge of fuel leaving the pump to get to the spray valve, and generally the passages in this valve are large enough to hold several charges. The fuel in most designs of spray valve is arranged to flow out into a sort of honeycombed space formed by a series of perforated disks one over another, separated by shoulders or spacers, permitting the fluid to spread over a large surface. The rush of air when the valve is opened picks up this oil. The timing of ignition—that is, the admission of the mixture to the cylinder—is therefore determined by the air. The spray valve simply admits air, and the sudden inrush of air at a pressure of around 1000 pounds, blows the fuel into the cylinder, which it enters as a very fine spray. This mixture is very cold at that point because it has been compressed to about 1000 pounds per square inch and allowed to cool. The resulting re-expansion carries its temperature away down. This mist of finely divided fuel and air is brought into the combustion chamber of the engine, where the temperature has been raised up to 1100 degrees F. by extremely high compression of the air charge, so whatever fuel is brought in is bound to burn, as there is a large excess of air present.

CHAIRMAN: Then it is not a question of getting an exact mixture, as it is in a gas engine?

MR. ARTHUR B. LAKEY: No, but if they want smooth running there should be no abrupt change in the supply from stroke to stroke. If there is an air pocket in, say, the center hole at the end of the plunger in the pump I showed, it could constitute a cavity which could easily contain enough air, at 1000 pounds pressure per square inch, to take up the whole displacement of that

pump when this air expands to atmospheric pressure during the suction stroke. Consequently even a small air pocket would make the fuel supply very erratic or eliminate it entirely while running at light load. Air pockets in the pipe-lines or pump body are similarly prejudicial.

Mr. Frank L. Egan: If I remember rightly, even at full load you have a great excess of air in the cylinder over what you actually need for combustion. That would give perfect combustion, and we find that by an adjustment of the pump for the admission of oil, so as to delay the point of admission a trifle, we get what you would call a "fatter" indicator card. You do not get an explosion such as you do with a natural-gas engine, but by delaying the admission you get more indicator card and reduce the heat. So it seems to slow the combustion down a little.

Mr. Arthur B. Lakey: The diagram looks a good deal like a Corliss card.

Mr. Frank L. Egan: You can by adjustment get a very pretty card.

Mr. Arthur B. Lakey: Fig. 12 shows types of indicator cards which might easily be obtained by burning identical charges of fuel by the Diesel (constant pressure) and the Otto (constant volume) methods.

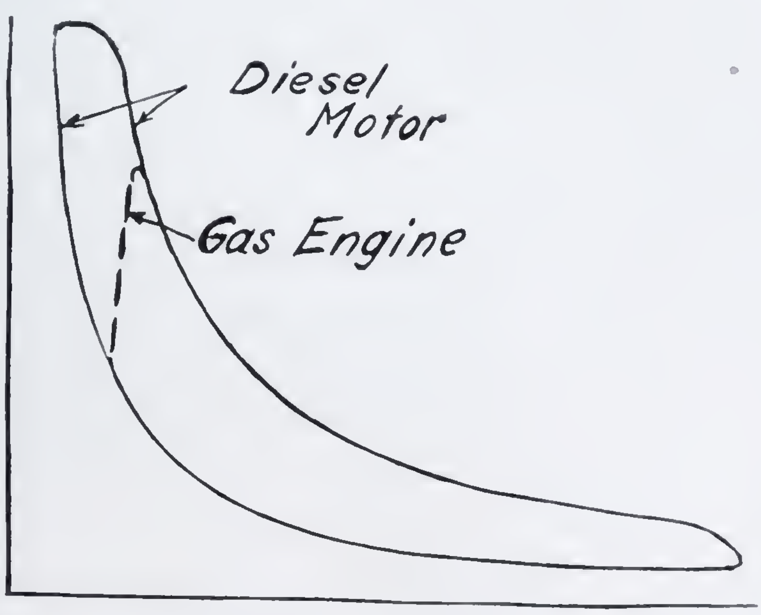

Fig. 12. Comparison of Indicator Cards.

A comparison of the areas of the two cards shows, incidentally, how so much more work is obtained from each working stroke with the Diesel engine than with the same amount of fuel with gasoline or other vaporized fuel, or with gas.

These engines can run on tar. They can even run on molasses for a while. The first Diesel engine used pulverized coal. That was the original object, but it was found that the blast of coal dust acted like a sand-blast on the piston, and the company which developed the earliest practical Diesel engine abandoned pulverized coal and adopted oil. In Germany they are pretty hard put to it for fuel, and tar oil is principally used; stuff that is not a success in any other form of combustion engine. Of course a Diesel engine could be run on gasoline, if there were any object in it.

Mr. Frank L. Egan: Would there not be additional difficulty in handling very small quantities of gasoline, in reducing the number of drops?

Mr. Arthur B. Lakey: Yes. I once counted the number of drops with a certain kind of oil and found there were 250 drops to the cubic inch. There would be a greater number for gasoline, of course. It depends on the surface tension, density, viscosity, etc. It will be understood, of course, that we have referred to the number of drops per charge simply to emphasize the minute quantities of fuel to be handled. Actually, the fuel has no opportunity to break up into drops, after leaving the pump, until it is picked up as very fine drops by the violent flow of air on its way to the cylinder.

The particular portion of oil which is expressed from the pump cylinder, determined in quantity by the governing mechanism in accordance with the instantaneous conditions of loading, will be unable to reach the cylinder for many strokes to come. It, however, causes an equal amount to leave the far end of the passage and flow over the atomizing surface in the spray valve. If the action of the pump were free from accidental variations, and the oil passages were entirely free from elastic deformation, the regulation would be all that could be desired.

THE CAISSON METHOD FOR FOUNDATIONS AND MINE SHAFTS

By George R. Johnson*

OUTLINE

1. Introductory
2. Building Foundations
    - Conditions in New York City
    - The Caisson
    - Caisson Disease
    - Examples
3. Bridge Piers
    - Historical
    - Types of Caisson
    - Examples
4. Mine Shafts
    - Historical
    - Type of Caisson Determined by Conditions
    - Examples

1. INTRODUCTORY

So great and varied has been the development of the caisson method within the past 20 years and so many sided are the problems to be met and solved in each particular case that to attempt a highly technical discussion of the subject would require a much more elaborate treatment than is intended in this paper. The matter presented here will follow along the lines of a general review, with numerous examples to illustrate the salient points, and will be treated under the following heads:

Building foundations.
Bridge piers.
Mine shafts.

*District Manager, The Foundation Co., Pittsburgh.

2. BUILDING FOUNDATIONS

CONDITIONS IN NEW YORK CITY

Doubtless no better examples of this class of foundation can be found anywhere than those obtaining in the lower Manhattan section of New York City where the characteristic geology imposes severe conditions necessitating the use of the pneumatic method.

In many places the soil for a considerable depth is of a very treacherous nature as layers of quicksand of extreme fluidity are encountered to a maximum thickness of 60 feet. The upper layers are usually filled ground and at a depth of from 10 to 30 feet below curb is the level of ground water, this being substantially the same elevation as mean tide in New York Bay. Below this the different strata of sand, gravel and quicksand vary greatly at different points. Just overlying the rock, which occurs at depths of from 50 to 150 feet, is usually a layer of about 10 feet of hard-pan which closely resembles concrete. Due to the extremely fluid condition of the quicksand it is very dangerous to attempt pumping in the open coffer-dam for the probable effect would be to draw the quicksand from under some near-by building and wreck it.

Before this section of Manhattan became so congested, comparatively little trouble was experienced in securing the proper foundation conditions near the surface, and the ordinary spread footing sufficed, but, because of the demand for more room which meant extremely high buildings and consequently tremendous foundation loads, it became imperative that the foundations be carried to hard-pan or rock, which meant in many cases a depth of 100 feet or more below street level; hence the pneumatic caisson.

In the old spread-footing type of foundation, a loading of four tons per square foot is common on compact sand and in the pneumatic caisson it ranges from 12 to 18 tons per square foot on the bottom, depending on the nature of the strata.

THE CAISSON

The pneumatic caisson is simply a bottomless box of cylindrical or rectangular section in plan, conforming to the section of the foundation pier. This box portion is called the working chamber, and the lower edges that are forced through the ground are called the cutting edges. On top of this working chamber is built a shaft of concrete, of the same sectional form, which is added to vertically as the sinking progresses, keeping the top above ground until the cutting edge reaches the desired depth, when the top will be at the proper elevation. Up through the center of the caisson, connecting the working chamber with the surface of the ground and built into the concrete, are one or more steel shafts surmounted by air locks for the passage of men and material (Fig. 1).

Due to its weight, the caisson sinks into the ground and the material that is forced into the working chamber is excavated. When water is reached, compressed air is pumped into the working chamber to a pressure sufficient to keep the water out and balance the head produced thereby. The material is now excavated by workmen in the compressed air and it is necessary to pass all the material through the air locks. This is done by means of a bucket a little smaller than the size of the shaft. If the weight of the caisson is not enough to sink it, temporary weight must be added to make up the difference This is usually in the form of pig-iron or else blocks of cast-iron made up for the purpose. Fig. 2 is a good example of a light caisson loaded with iron blocks. Great care is taken to maintain the caisson vertically. This is accomplished by loading it eccentrically or else by digging under the high edge, or by both methods in case it should get out of plumb. When the desired depth is reached, the bottom is cleaned off and the working chamber and shafts are concreted, forming a solid pier.

With the arrangement described, it is readily seen that no appreciable disturbance of the surrounding material need occur, thus permitting the caisson to be placed up against the footing of adjoining buildings. Proper precautions against settlement of such footings are always observed.

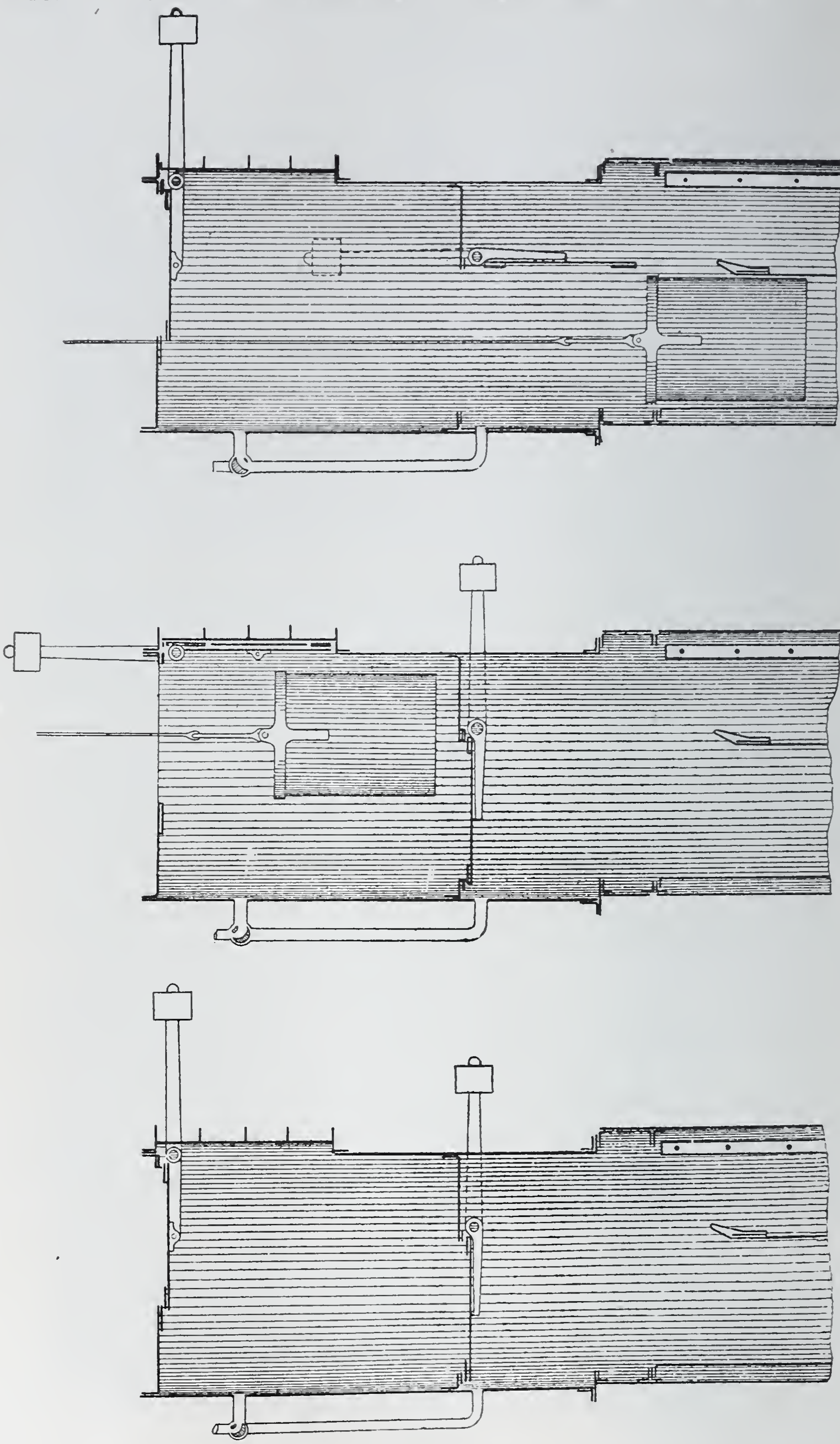

Fig. 1. Moran Air Lock.

Fig. 2. Bryant Building Foundations. Caisson Loaded to Secure Necessary Sinking Weight.

CAISSON DISEASE

The matter of caisson disease or "bends" will be touched upon briefly. In the early days of the caisson, the effect of compressed air on the body was little understood and frequent occurence of bends was the result. However, it is now generally known among air men that if the following precautions are observed there is scarcely any danger from this source.

1. Selection of men by physical examination.
2. Proper ventilation of working chamber.

3. A graduated working time, depending upon the air pressure.
4. A slow rate of decompression.
5. Proper facilities for caring for incipient cases.

Extracts from the New York State law are appended to this paper.

It is interesting to note that in the three pneumatic caissons of Pennsylvania Railroad Bridge no. 122 at Rochester, Pa., just being completed, not a single case of "bends" occurred.

EXAMPLES

Singer Building. The foundations for this job consist of 30 pneumatic caissons sunk to a depth of about 70 feet below curb, 20 of these being under the tower, and designed to carry a load of 15 tons per square foot on the rock (Fig. 3).

After caisson 30–43 under the tower had been sunk to hard-pan and the working chamber and shaft concreted, it was decided to carry the tower caissons to rock. This meant underpinning the caisson and this was accomplished by sinking an adjacent caisson, 29–35, to hard-pan and tunneling under the one to be underpinned, excavating under the same to rock and filling with concrete. All this was done without an accident and this is probably the only case in which a caisson has been underpinned.

Municipal Building. The foundations of this building consist of 106 pneumatic caissons sunk through water, quicksand, coarse sand and gravel to the maximum depth of 112 feet below ground-water level, a record up to that time of construction.

Because the rock in the northern part of the lot dropped off to a prohibitive depth, 38 caissons in this part were carried to firm sand, 77 feet below street level, and designed for six tons per square foot. The remaining 68 were carried to rock and designed for 15 tons per square foot. This is clearly shown in Fig 4.

Since Chambers Street passes across the middle of the lot it was necessary to provide for heavy street traffic during the entire construction. See Fig. 5.

Woolworth Building. This building rests on 66 piers varying in size from 6 feet 6 inches, to 21 feet in diameter, sunk as

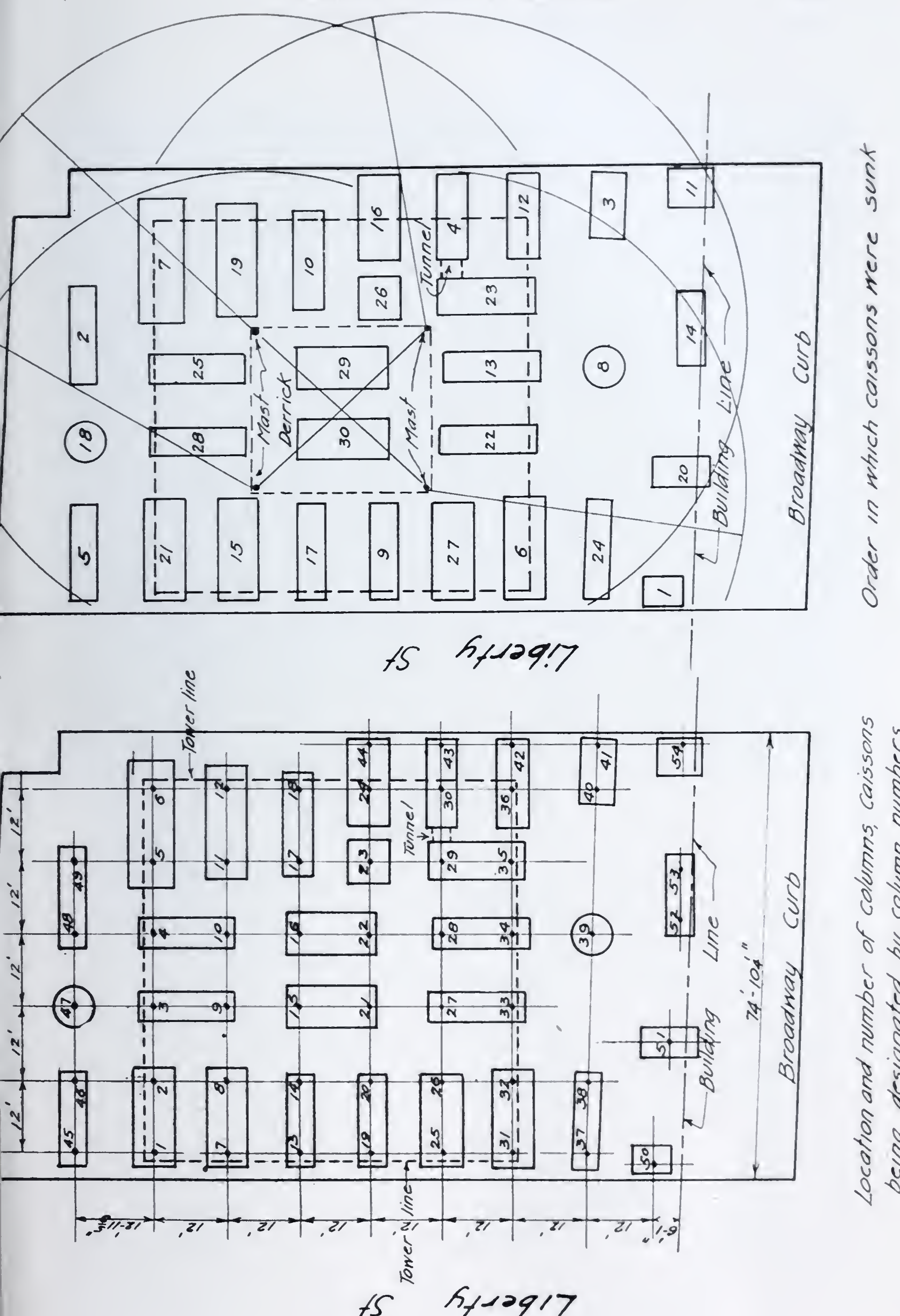

Fig. 3. Caisson Plan for Singer Building.

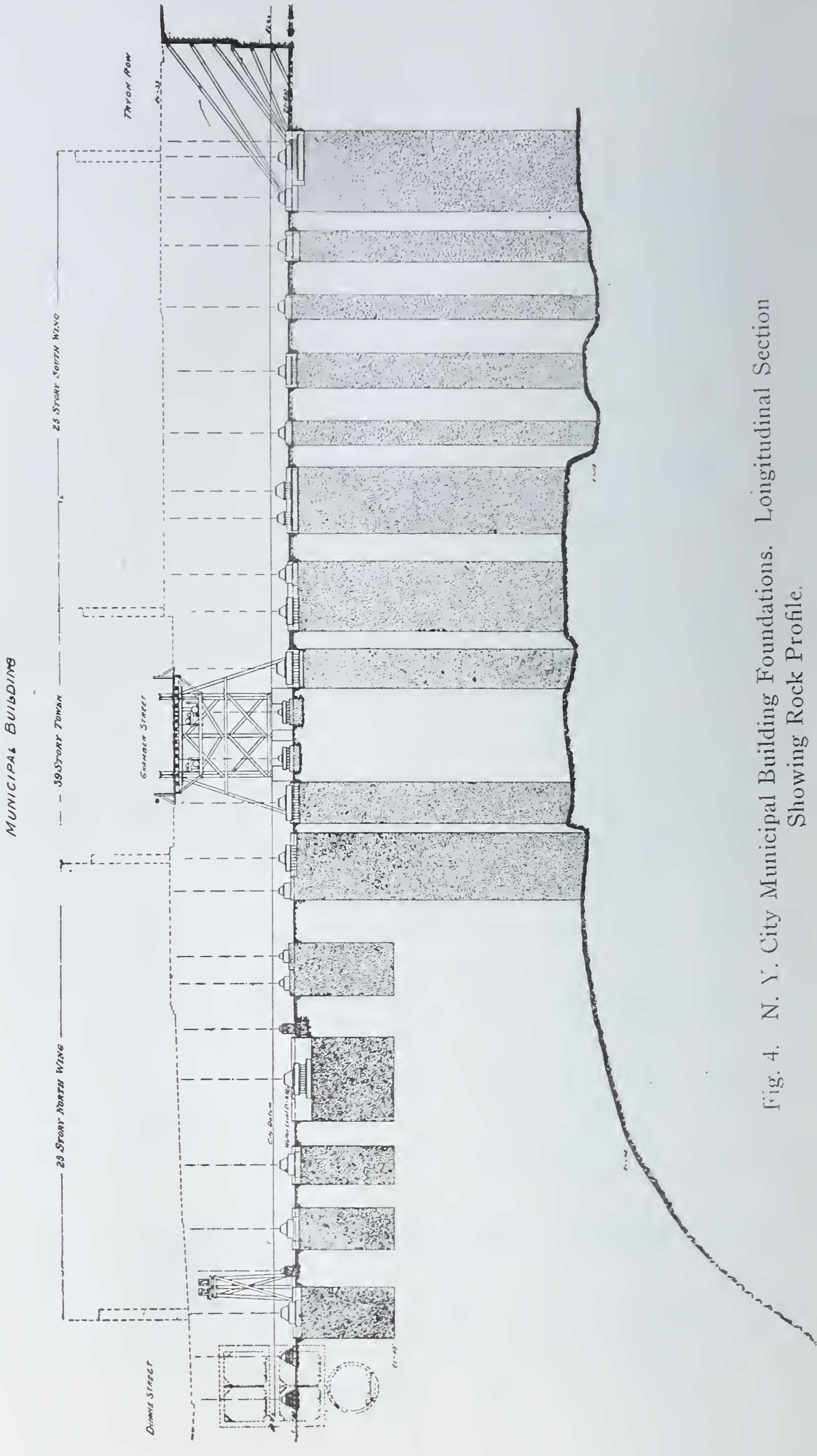

Fig. 4. N. Y. City Municipal Building Foundations. Longitudinal Section Showing Rock Profile.

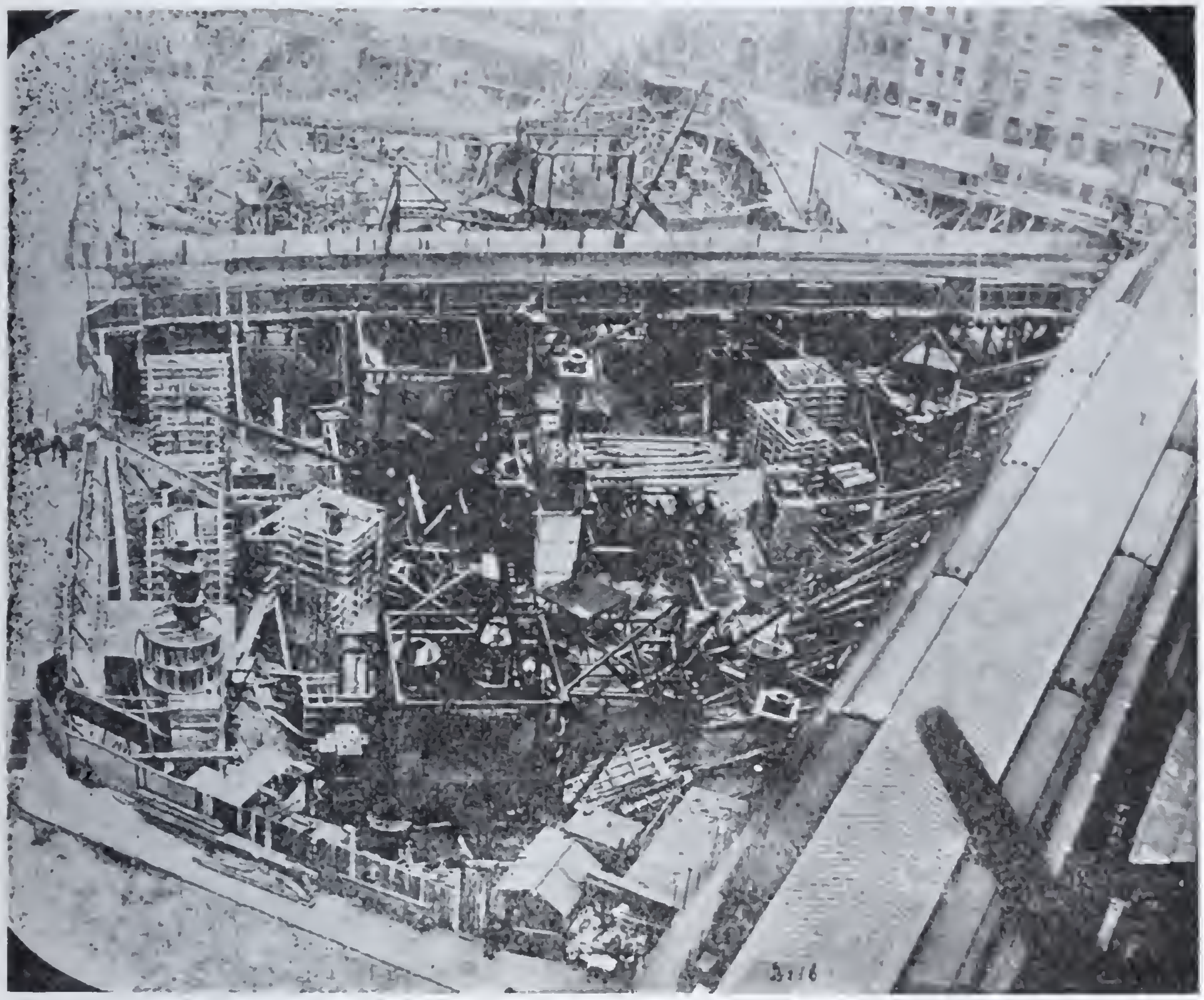

Fig. 5. N. Y. City Municipal Building. View During Sinking Period.

pneumatic caissons to rock a maximum of 120 feet below street level. The most notable thing about these caissons is the fact that they carry the highest building in the world; it being 750 feet above the curb or a maximum of 870 feet above the base of the deepest caisson.

The above covers a few of the most important pneumatic jobs in lower Manhattan and it may be said that the present sky-line of this section exists by virtue of this type of foundation.

3. BRIDGE PIERS

HISTORICAL

Up to 1859 only two bridges had been built in the United States on pneumatic pile foundations; one across the Pedee and the other across the Santee River in South Carolina.

In 1869 the first pneumatic pier foundations west of the Allegheny Mountains were put down at Omaha in the Missouri River. The piers for the St. Louis Eads Bridge and Roeblings East River Bridge were sunk between 1870 and 1873.

TYPES OF CAISSON

In general it may be said that caissons for bridge piers are divided into two types—caissons for land piers which are built up and sunk in position, and caissons for river piers built up on shore, launched like a boat, towed to position and sunk. In building the second type, the matter of buoyancy must be considered and the caisson must be so designed that when launched it will float upright. In order to hold a river caisson in position against the force of the current and maintain it in proper alignment, guide piles are usually driven around it. See Fig. 13.

EXAMPLES

Clinton Bridge. This structure was built by the Chicago and North Western Railway, over the Mississippi River at Clinton, Iowa. It consists of 22 cylinder piers dredged down in the open, 11 piers and abutments constructed by the coffer-dam method and three piers sunk by the pneumatic method (Fig. 6). The last named range in depth from 40 to 70 feet.

Red River Bridge No. 1. This work at Winnipeg, Canada, consisted of extending the old piers for the purpose of double tracking, and illustrates a very interesting piece of foundation work. The old pier rested on hard-pan but as it had been in so long it was decided that it had its full settlement and the new portion should be carried to rock. This is clearly seen in Fig. 7 (folding plate).

Harrison River Bridge. The only exceptional feature of this job at Harrison Mills, B. C., was the caissons for pier no. 8 (Fig 8). This double caisson was used because of the extreme slope of the rock surface.

Mud Lake Bridge. Just a word in regard to this bridge substructure. Fig. 9 clearly shows the nature of the material and the conditions which necessitated the deepest bridge caisson in Canada to that date—103 feet 8 inches below water level. This bridge is located at Perth, Ontario.

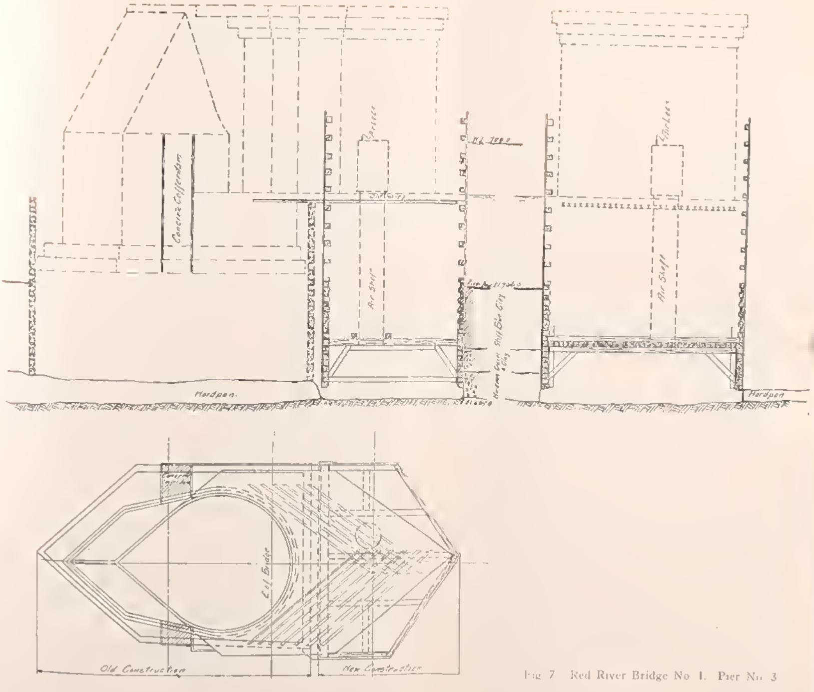

Fig 7 Red River Bridge No 1. Pier No. 3

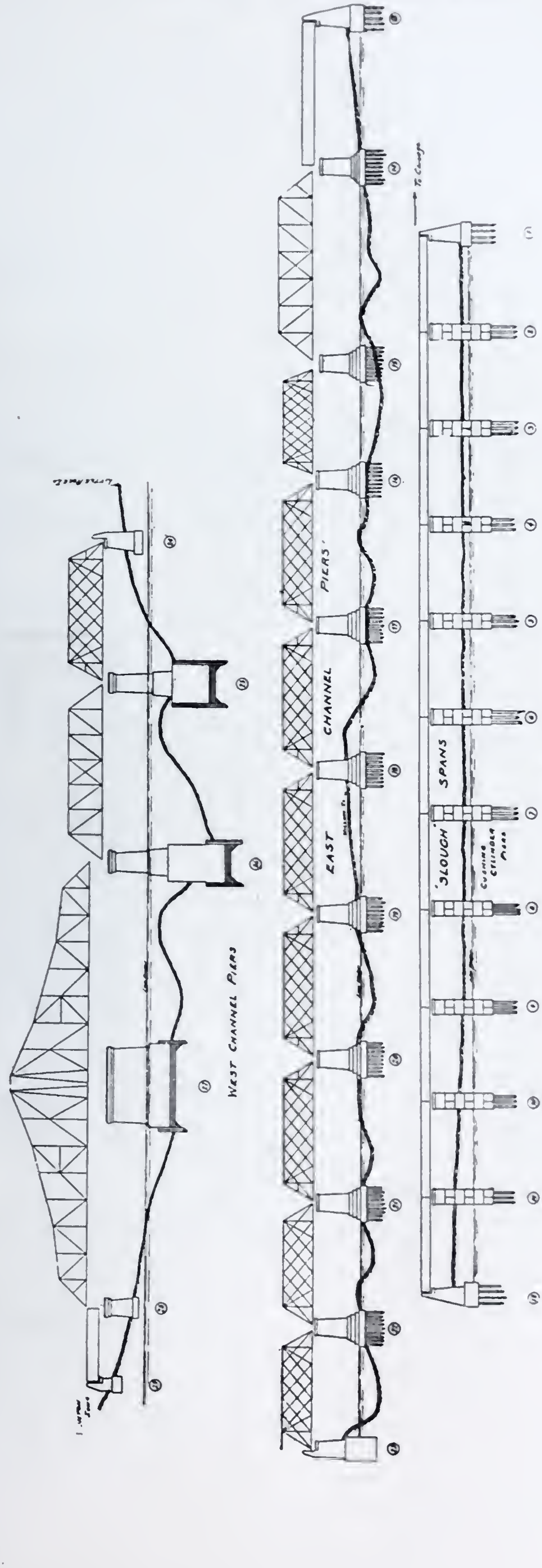

Fig. 6. Clinton Bridge. General Elevation Showing All Piers.

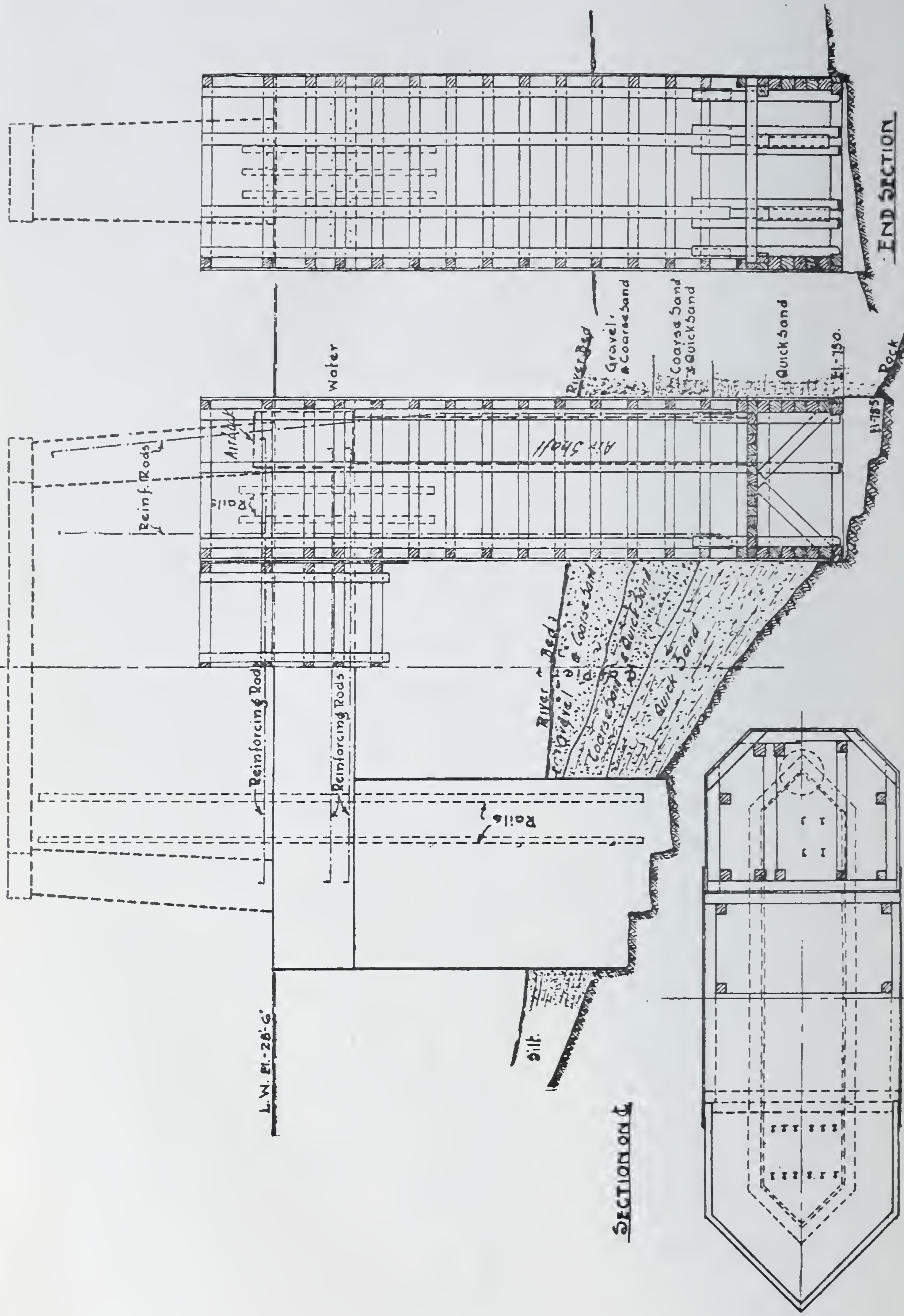

Fig. 8. Harrison River Bridge. Typical Caisson.

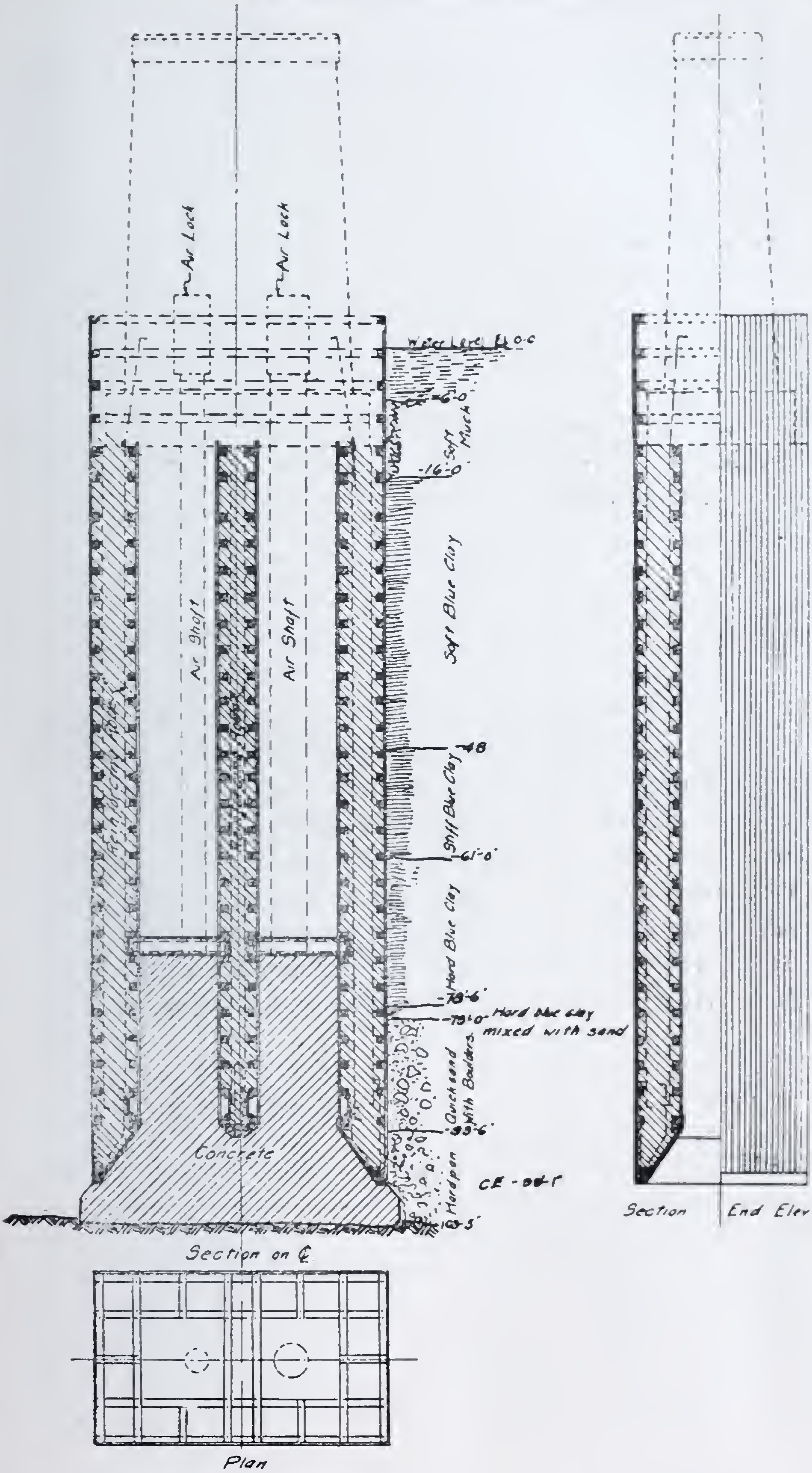

Fig. 9. Mud Lake Bridge. Typical Caisson.

Pennsylvania Railroad Bridge No. 122. The work for this bridge over the Beaver River at Rochester, Pa., is at present nearing completion. It consists of two land caissons, 28 by 45 feet and 19 by 47 feet, and one river caisson 18 feet 6 inches by 57 feet 6 inches. Although the caissons are not unusually deep, the water and ice conditions were very severe during the greater part of the sinking period, keeping the whole caisson completely submerged for a considerable length of time, but not stopping the work—another point in favor of pneumatic work. See Fig. 10–17.

The rock line is about 35 feet below pool level but for a considerable length of time the river stage was 20 feet of water, though the normal depth is six feet.

4. MINE SHAFTS

HISTORICAL

Prior to 1890 few, if any, mine shafts on this continent had been constructed through water-bearing soils or fine sands to a greater depth than 50 feet below ground-water level. Only inexpensive drop shafts of timber had been attempted.

The construction of deep shafts through water-bearing soils must meet the following requirements: The first cost must be reasonable; the shaft must be strong enough to withstand the pressure to which it is subjected; the maintenance cost must be low; the shaft must be plumb; it must be waterproof and fireproof.

TYPE OF CAISSON DETERMINED RY CONDITIONS

If the soils are water bearing but firm, it is often possible to sink an open dredged caisson to rock, then pump the water down and seal the cutting edge into the rock. This procedure was followed in the shaft just being completed at Vesta Mine no. 7, West Brownsville, Pa.

It is not always possible to pump down, nor are the conditions always such as to permit of dredging, in which case a pneumatic caisson is the method used. More frequently, how-

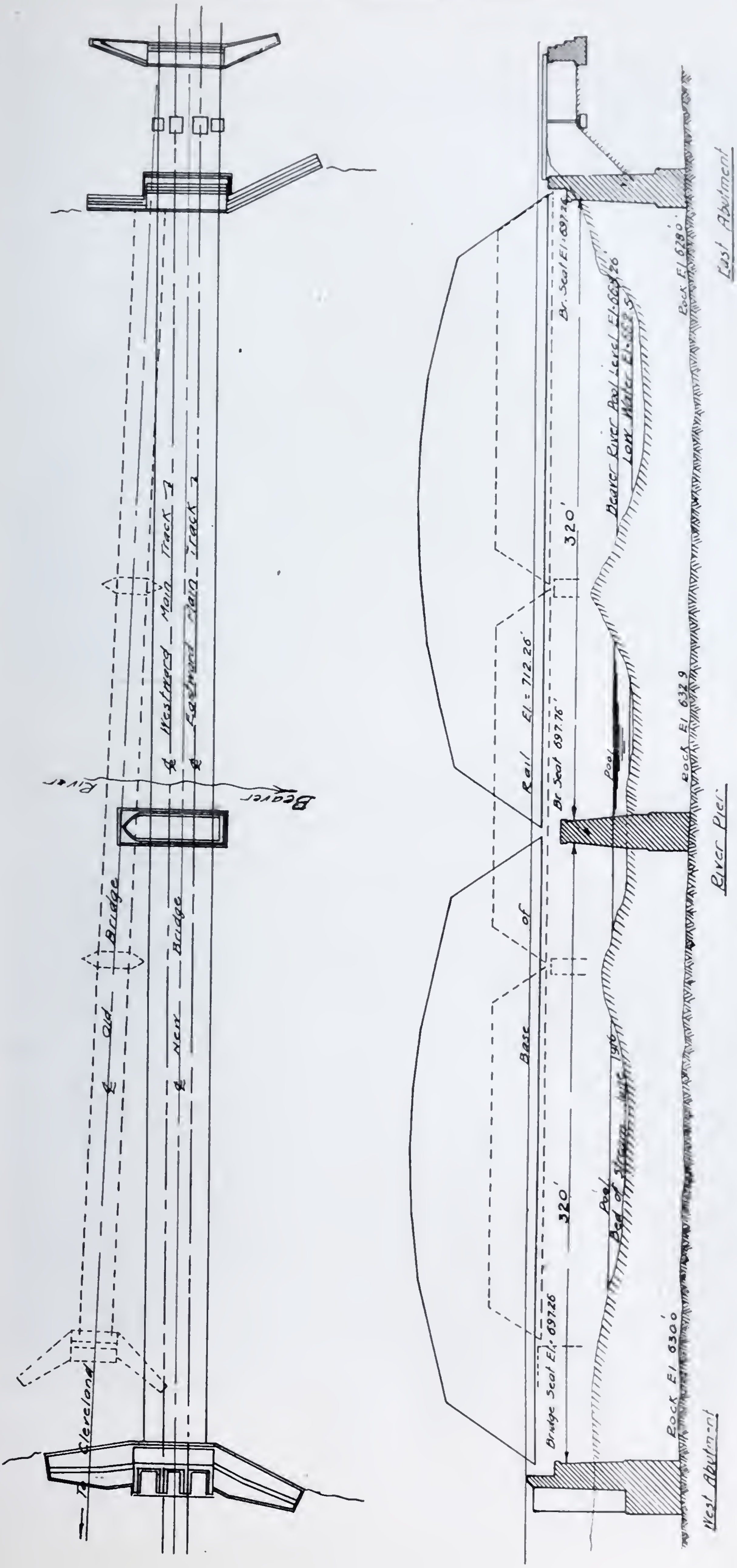

Fig. 10. Beaver River Bridge, Rochester, Pa.

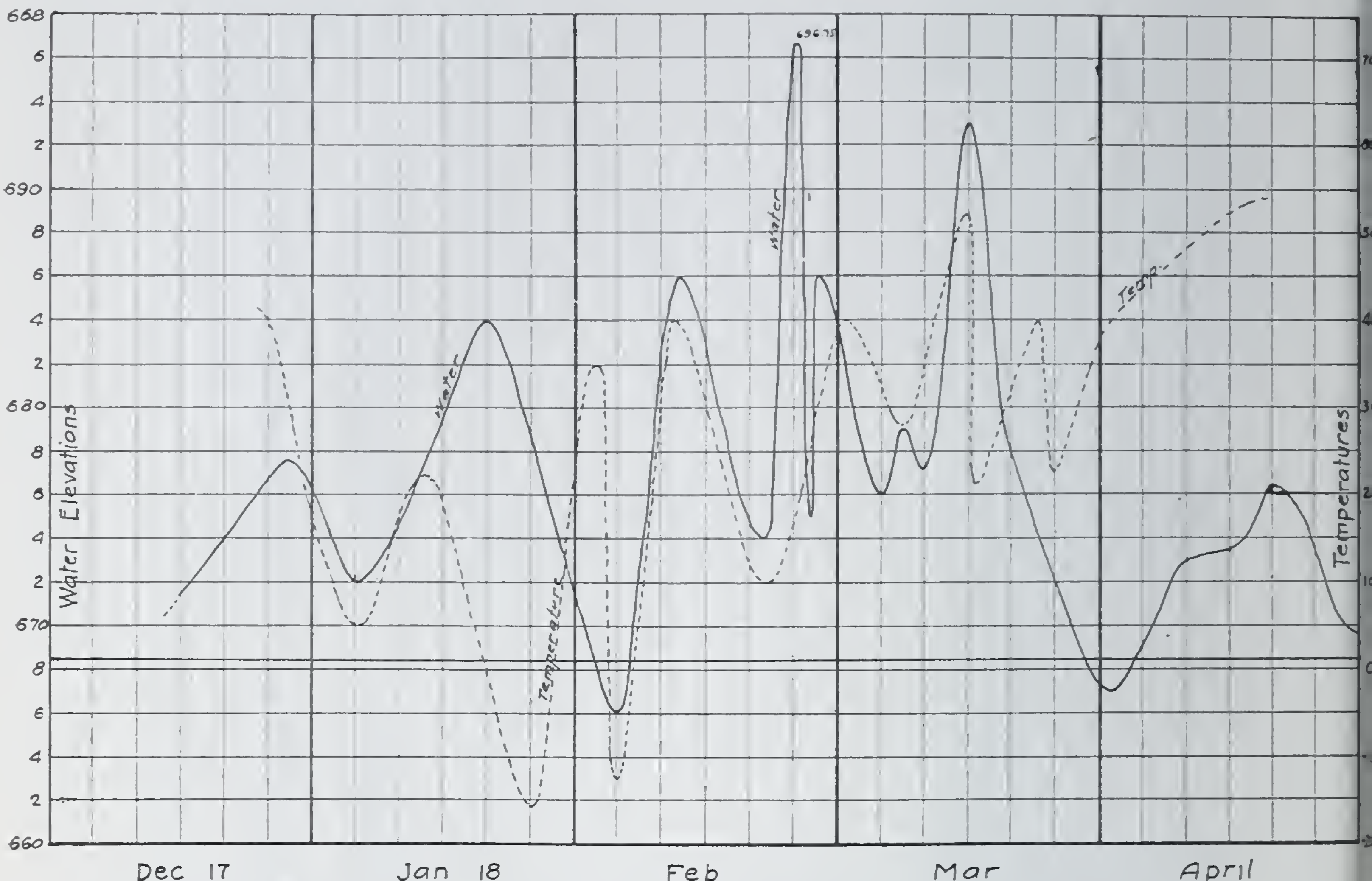

Fig. 11. Temperature and Water Variations, Beaver River, Rochester, Pa.

Fig. 12. Beaver River Bridge. River Caisson on Ways Ready for Launching.

Fig. 13. Beaver River Bridge. Sinking River Caisson in 18 Feet of Water.

Fig. 14. Beaver River Bridge. East Abutment. Caisson with Air on, Weighted. Concreting River Pier in Background.

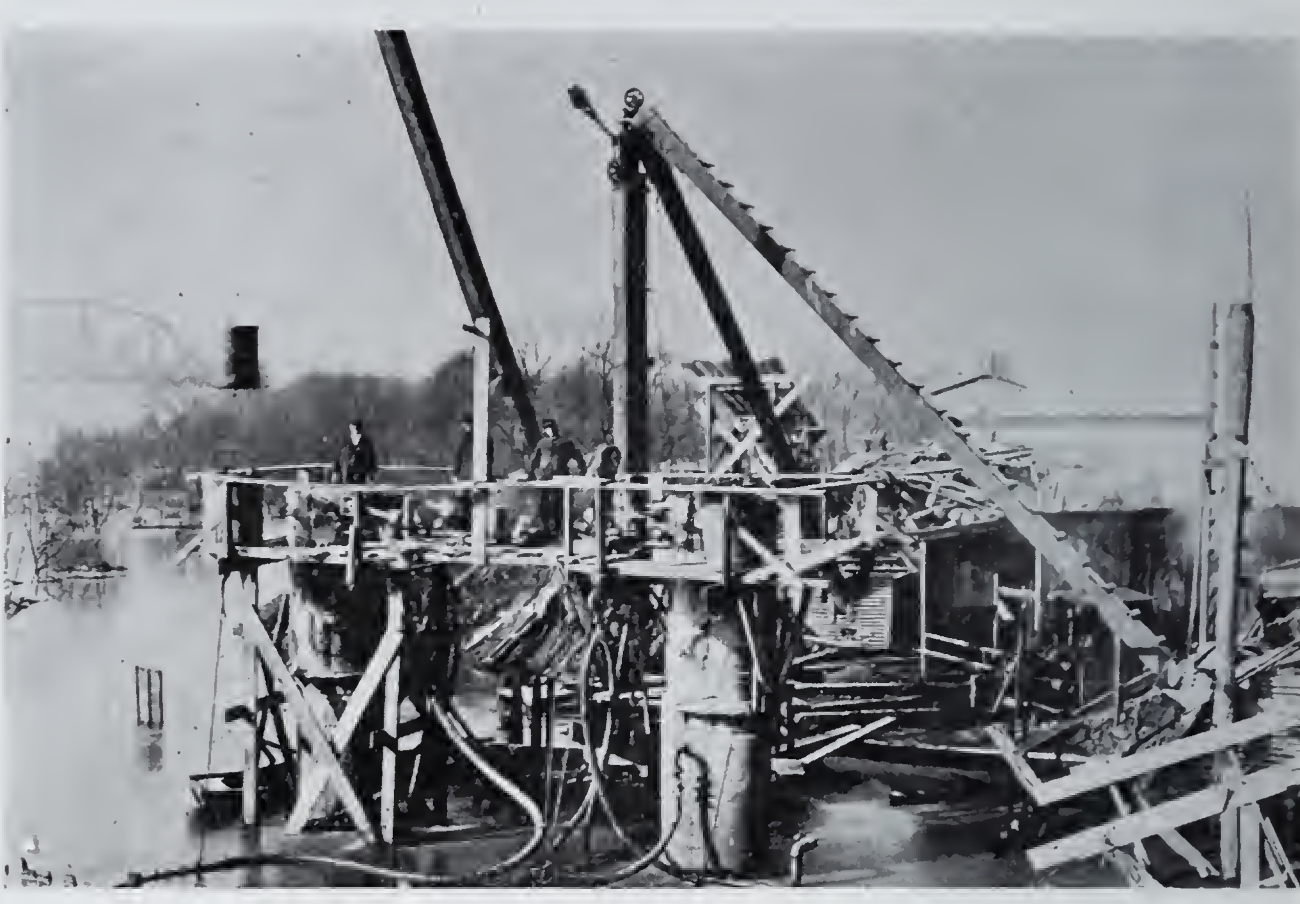

Fig. 15. Beaver River Bridge. West Abutment. Caisson with Air on; 28 Feet of Water.

Fig. 16. Beaver River Bridge. High Water, Feb. 21, 1918.

Fig. 17. Beaver River Bridge. Ice Conditions, Feb. 6, 1918.

ever, a combination of the two types represents the best solution, both from an engineering and an economic point of view. In this case the caisson is dredged down to rock, where a deck is placed near the bottom, air applied to drive the water out and the seal placed under air, after which the deck is removed and the shaft completed through the rock in the open.

EXAMPLES

Morton Shaft No. 5. This shaft was constructed for the Tod-Stambaugh Company, at Hibbing, Minn. The depth to rock was 190 feet with a water head of 165 feet. The first attempt was by means of a single drop shaft which after twelve months of work was carried down 170 feet where it stuck and resisted all further efforts to move it (Fig. 18). This shaft was finally completed the remaining 20 feet to rock by a combination of the pneumatic method and pumping. A timber deck, loosely fitting the inside of the shaft, was built and floated down by pumping out until within 10 feet of the cutting edge where a recess was cut all around the wall as a keyway. On the top of the timber

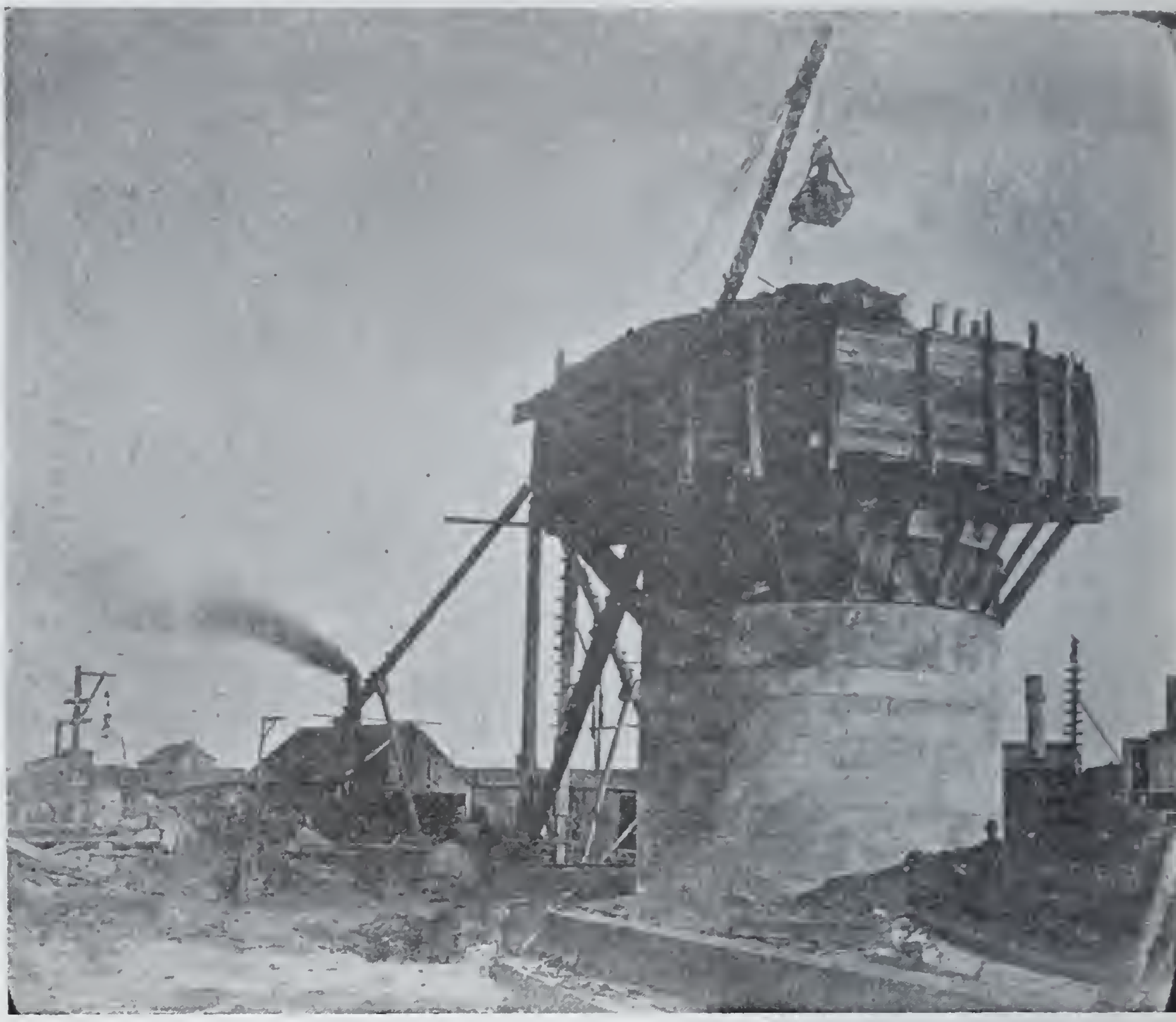

Fig. 18. Morton Shaft No. 5. Platform for Weighting Shaft During Sinking Period.

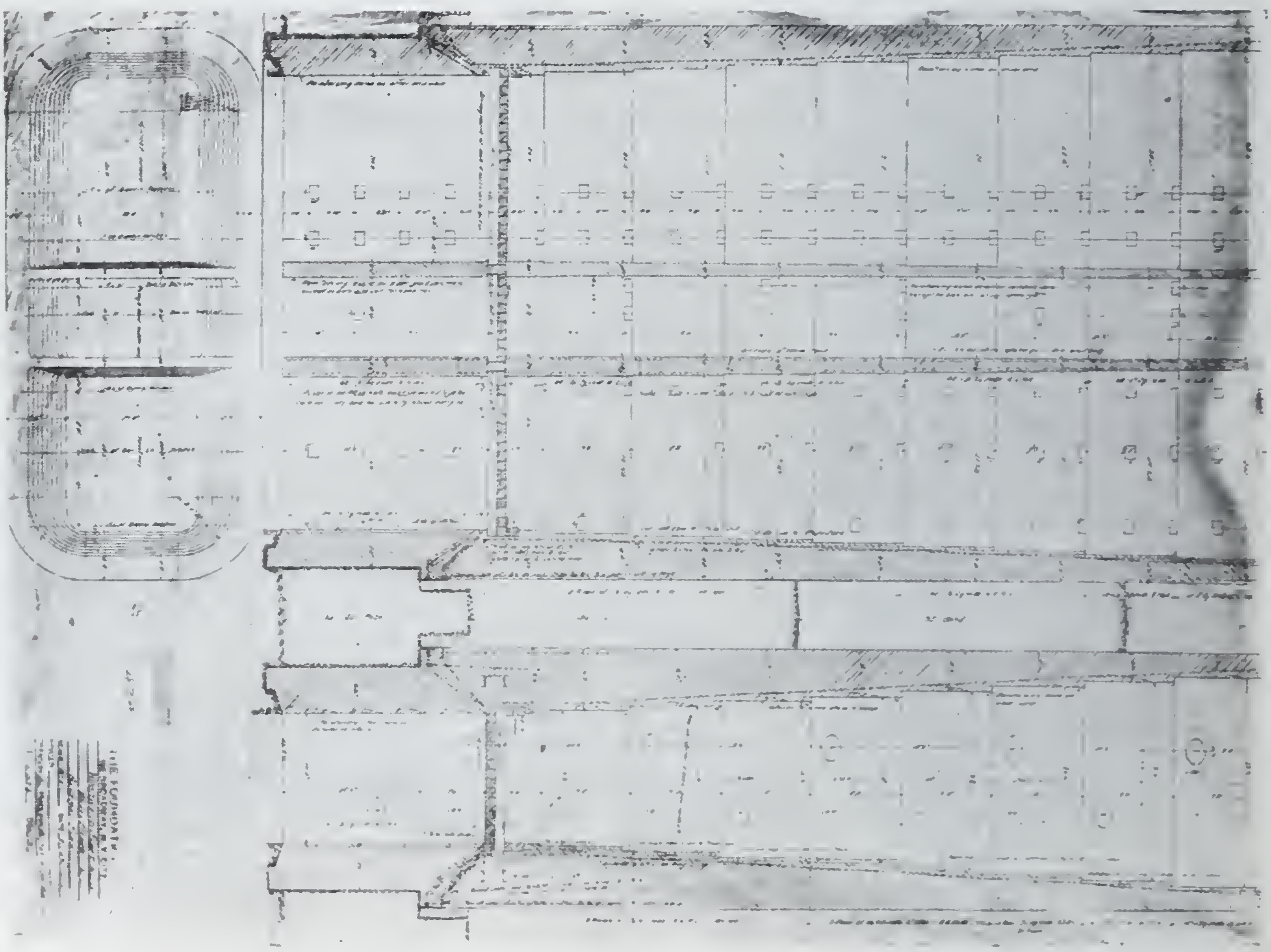

Fig. 19. Woodward Shaft Sections.

Fig. 20. Woodward Shaft. Cutting Edge.

deck, and bonded into the notch, was deposited a concrete seal 12 feet thick. When the concrete had set, holes were drilled through the sides of the shaft just above the concrete deck and valves placed therein. These were then opened and the shaft converted into a sump. After pumping for several weeks a large basin was drained all around the shaft and kept dry, when air was applied below the deck and the shaft underpinned the remaining 20 feet to rock. Without pumping, the air pressure necessary with 165 feet of water would have been prohibitive.

Woodward Shaft. This shaft was built for the Delaware, Lackawanna & Western Railroad, at Wilkes-Barre, Pa., 28 by 60 feet and 70 feet deep—the largest shaft of its kind constructed up to that date. It consisted of three compartments—a ventilating compartment, a manway, and a pump compartment. See Fig. 19–20.

Vesta Shaft No. 7. There is at present being completed for the Vesta Coal Company at West Brownsville, Pa., a reinforced

concrete drop shaft of the open dredging type, 19 feet 6 inches by 33 feet and about 50 feet deep. This caisson was sunk to rock and sealed in 20 days.

This completes the treatment of the subject as intended in this paper and, though many other interesting cases could be cited, the ones given are representative and it is hoped that they have proven of interest.

APPENDIX

EXTRACT FROM NEW YORK STATE COMPRESSED AIR LAW*

Section 134a. Hours of Labor. All work in the prosecution of which tunnels, caissons or other apparatus or means in which compressed air is employed or used shall be conducted subject to the following restrictions and regulations: When the air pressure in any compartment, caisson, tunnel or place in which men are employed is greater than normal and shall not exceed 21 pounds per square inch, no employee shall be permitted to work or shall remain therein more than eight hours in any 24 hours and shall only be permitted to work under such air pressure provided he shall, during said period, return to the open air for an interval of at least 30 consecutive minutes, which interval his employer shall provide for. When the air pressure in any compartment, caisson, tunnel or place in which men are employed is greater than normal and shall equal 22 pounds and does not exceed 30 pounds per square inch, no employee shall be permitted to work or remain therein more than six hours, such six hours to be divided into two periods of three hours each with an interval of at least one hour between each such period. When the air pressure in any such compartment, caisson, tunnel or place shall exceed 30 pounds and shall not equal 35 pounds per square inch, no employee shall be permitted to work or remain therein more than four hours, such four hours to be divided into two periods of two hours each, with an interval of at least two hours between each such period. When the air pressure in any such compartment, caisson, tunnel or place shall equal 35 pounds and shall not exceed 40 pounds per square inch, no such employee shall be permitted to work or remain therein more than three hours in any 24 hours, such three hours to be divided into periods of not more than one and one-half hours each, with an interval of at least three hours between each such period; when the air pressure in any such compartment, caisson, tunnel or place shall equal 40 pounds and shall not equal 45 pounds per square inch, no employee shall be permitted to work or remain therein more than two hours in any 24 hours, such two hours to be divided into periods of not more than one hour each, with an interval of at least four hours between each such period; when the air pressure in any such compartment, caisson, tunnel or place shall equal 45 pounds per square inch and shall not exceed 50 pounds per square inch, no employee shall be permitted to work or remain there more than 90 minutes in any 24 hours and such 90 minutes to be divided

*Quoted from Charles Evan Fowler's "Practical Treatise on Sub-Aqueous Foundations." Ed. 3, 1914. pp. 231–234. New York: Wiley.

into periods of 45 minutes each, with an interval of not less than five hours between each such period; no employee shall be permitted to work in any compartment, caisson, tunnel or place where the pressure shall exceed 50 pounds per square inch, except in case of emergency. No person employed in work in compressed air shall be permitted by his employer or by the person in charge of said work to pass from the place in which the work is being done to atmosphere of normal pressure, without passing through an intermediate lock or stage of decompression, which said decompression shall be, where the work is being done in tunnels, at the rate of three pounds every two minutes unless the pressure shall be over 36 pounds, in which event the decompression shall be at the rate of one pound per minute; and which said decompression shall be, where the work is being done in caissons, at the following rates:

Where the pressure is not over 10 pounds per square inch the time of decompression shall be one minute; when the pressure is over 10 pounds, but does not exceed 15 pounds, the time of decompression shall be two minutes; when the pressure is over 15 pounds, but does not exceed 20 pounds, the time of the decompression shall be five minutes; when pressure is over 20 pounds, but does not exceed 25 pounds, the time of decompression shall be 10 minutes; when pressure is over 25 pounds, but does not exceed 30 pounds, the time of decompression shall be 12 minutes; when pressure is over 30 pounds, but does not exceed 36 pounds, the time of decompression shall be 15 minutes; when pressure is over 36 pounds, but does not exceed 40 pounds, the time of decompression shall be 20 minutes; when pressure is over 40 pounds, but does not exceed 50 pounds the time of decompression shall be 25 minutes.

All necessary instruments shall be attached to all caissons and airlocks showing the actual air pressure to which men employed therein are subjected, and which instruments shall be accessible to and in charge of a competent person who shall not be employed more than eight hours in any 24 hours.

Section 134b. Medical Attendance and Regulations. Any person or corporation carrying on any tunnel, caisson or other work in prosecution of which men are employed or permitted to work in compressed air, shall, while such men are so employed, also employ and keep in employment, one or more duly qualified persons to act as medical officer or officers who shall be in attendance at all necessary times while such work is in progress, and whose duty it shall be to administer and strictly enforce the following:

a. No person shall be permitted to work in compressed air until after he shall have been examined by such medical officer and reported by such officer to the person in charge thereof as found to be qualified, physically, to engage in such work.

b. In the event of absence from work, by an employee for 10 or more successive days for any cause, he shall not resume work until he

shall have been re-examined by the medical officer and his physical condition reported, as hitherto provided, to be such as to permit him to work in compressed air.

c. No person known to be addicted to the excessive use of intoxicants shall be permitted to work in compressed air.

d. No person not having previously worked in compressed air shall be permitted during the first 24 hours of his employment to work for longer than one-half a day period as provided in Section 134a; and after so working shall be re-examined and not permitted to work in a place where the pressure is in excess of 15 pounds unless his physical condition be reported by the medical officer, as heretofore provided, to be such as to qualify him for such work.

e. After a person has been employed continuously in compressed air for a period of three months he shall be re-examined by the medical officer and he shall not be allowed, permitted or compelled to work until such examination has been made and he has been reported, as heretofore provided, as physically qualified to engage in compressed air work.

f. The said medical officer shall at all times keep a complete and full record of examinations made by him, which record shall contain dates on which examinations were made and a clear and full description of the person examined, his age and physical condition at the time examined, also the statement as to the time such person has been engaged in like employment.

g. Properly heated, lighted and ventilated dressing rooms shall be provided for all employees in compressed air, which shall contain lockers and benches and shall be open and accessible to the men during the intermission between shifts. Such rooms shall be provided with baths, with hot- and cold-water service and a proper and sanitary toilet.

h. A medical lock shall be established and maintained in connection with all work in compressed air when the maximum pressure exceeds 17 pounds as herein provided. Such lock shall be kept properly heated, lighted and ventilated and shall contain proper medical and surgical equipment. Such lock shall be in charge of a certified trained nurse selected by the medical officer, who shall be qualified to render temporary relief.

i. Whenever in the prosecution of caisson work in which compressed air is employed, the working chamber is less than 10 feet in length and when such caissons are at any time suspended, or hung, while work is in progress, so that the bottom of the excavation is more than nine feet below the deck of the working chamber, a shield shall be erected in the working chamber for the protection of the workmen.

j. Whenever in the prosecution of work in which compressed air is employed, a shaft is used, all such shafts shall be provided with a safe, proper and suitable ladder for its entire length.

k. Wherever in the prosecution of work in tunnels, caissons or other apparatus or means in which compressed air is employed or used, lights other than electric lights are used, the said lights shall at all times be guarded.

l. All passage ways in work, wherein compressed air is employed or used, shall be kept clear and properly lighted.

DISCUSSION

Mr. W. A. Weldin:* It might be of interest to a good many of us to go a little more minutely into the details of the Vesta shaft. Is the concrete lining reinforced to resist the full hydrostatic head?

Mr. George R. Johnson: The reinforcement is carried from the cutting edge up the outside face as well as the inside face for about 12 to 15 feet. After that, there is just reinforcement enough to build up to bind it to the next section as it goes down. This caisson was carried down to the rock by dredging. When it got to the rock it was sealed to it, the rock was then taken out and the caisson lined, as ordinarily, for the remainder of the distance. It is designed to stand a flood stage.

Mr. W. A. Weldin: Water tight?

Mr. George R. Johnson: Yes. I do not mean that it would not sweat, but there is no leakage in the shaft.

Mr. W. A. Weldin: Is it a rectangular shaft?

Mr. George R. Johnson: The sides are parallel and the ends curved.

Mr. W. A. Weldin: Do you leave recesses in the lining for the buntons?

*Blum, Weldin & Co., Pittsburgh.

MR. GEORGE R. JOHNSON: There were recesses left—boxes for the buntons—square on one side and notched on the other side so the buntons could be dropped down into them.

MR. W. A. WELDIN: How thick were the walls?

MR. GEORGE R. JOHNSON: I do not happen to remember, but I should say, offhand, about four inches.

MR. C. T. DAY:* What were the proportions of the concrete used?

MR. GEORGE R. JOHNSON: Ordinarily 1:2½:5, but in the lower portion we put 1:2:4 to be safe. There is virtually where all the strain comes. If you hit a boulder, down there, all the strain would be taken up before getting very far up.

MR. W. E. SNYDER, *President*:† I did not quite understand your explanation as to why the water was overflowing.

MR. GEORGE R. JOHNSON: Suppose the caisson is passing through gravel, or something like that, that holds the shaft up. You excavate until the bottom of the excavation is a good way below the cutting edge. Then you expect the shaft to drop. Instead of dropping, the soil drops and rushes in around the cutting edge and, as the water level is above the bottom, it is pushed up like a piston and floods over the top.

MR. W. E. SNYDER: In the same way the bottom of the Panama Canal rises at times from outside pressure?

MR. GEORGE R. JOHNSON: Yes, in a way.

MR. W. E. SNYDER: How many men ordinarily work in a caisson?

*Detailer, American Bridge Co., Pittsburgh.

†Mechanical Engineer, American Steel and Wire Co., Pittsburgh.

Mr. George R. Johnson: That depends altogether on the size of the caisson. With a box 6 by 6 feet there are usually two men and a foreman. Due to union rules the foreman has to be there but he is merely a spectator. Two men actually do the work. It runs from that up to as many as 40 men in a big caisson.

Another means of getting material out is a blowpipe—a pipe with a valve at the bottom—the sand-hog lets in sand with a little water and the pressure shoots it up out of the pipe. It is pretty rough on the pipe. They use a regular blowpipe elbow that can be replaced when it is worn out and I have seen these elbows wear out in three hours.

I spoke of New York caissons. You have two types of caisson right here in Pittsburgh. The Oliver Building and the First National Bank Building were both pneumatic caisson jobs.

Mr. C. T. Day: I would like to ask how the cost of work of that kind is estimated.

Mr. George R. Johnson: We have kept exact cost records. How they originally started I do not know. We know what it costs to excavate up to certain pressures. The rules that New York State has put into her laws, prohibit working in air pressures above 20 pounds for a period of greater than eight hours. The men can work four hours in the air and then they must have half an hour out and then 3½ hours in again. When the pressure goes over 20 pounds the time is decreased. The next step is to six hours and the following one is to four hours. When you get up to 50 pounds of air the men work only 1½ hours—two three-quarter hour periods separated by an interval of five hours outside.

To estimate, you must know the depth of water. The price per yard will vary with the number of hours the men can work. It will run very uniformly up to 20 or 22 pounds. After that it increases very materially, and as the hours of work decrease the wages increase materially. When a sand-hog in the old days got $3 (when common labor was getting $1.50) for 1½ hours work he was a multimillionaire. I do not know what the union rules would call for now for 1½ hours work.

Working under compressed air there is plenty of oxygen and you would be surprised to see how quickly anything will burn. We use candles ordinarily and when you blow one out, if there is a little glow left in the wick, it will immediately break out in flame again; and if your clothes get on fire there is nothing to do but flap them in the water.

Mr. W. E. Snyder: What are some of the most noticeable sensations due to working under air pressure?

Mr. George R. Johnson: Put a lazy man under air and you get a good day's work out of him, as the oxygen is very invigorating. The danger does not occur in going into the lock. Passing through the Pennsylvania tubes to New York City you will get a quarter or a half pound pressure. The effect is felt in the ears and you instinctively swallow as a method of relieving the pressure. The real pain comes when you are entering the air, but the damage is done when you are coming out. You have to go in slowly to permit the air pressure to equalize what you have on the outside. Coming out you have to go through the reverse process and it takes a long time. I think the New York state law requires 1½ hours when coming out of 50 pounds pressure. If you come out too fast you get caisson disease or "bends," with sensations somewhat resembling rheumatism. If you have a bad case, go back in at about two-thirds the pressure at which you contracted the disease, and slowly decompress.

Mr. George H. Danforth:* Is there any attempt to give a little clearance to the body of the caisson?

Mr. George R. Johnson: No. That is an exploded theory. It was attempted in one of the first caissons that was sunk in New York. Instead of using concrete they used brick and constructed the caisson with a slight batter. The ground flows in and grips it and, as it flows in, it will undermine the adjoining buildings also. Fourteen or fifteen years ago when I started in this business it was stated that that was the proper method, but I haven't heard it even talked of in the last 10 years. It does not prevent the skin friction as you would think.

*Structural Engineer, Jones & Laughlin Steel Co., Pittsburgh.

Mr. George H. Danforth: I was thinking if it did not prevent it, it might at least ease it up a little.

Mr. George R. Johnson: It does not help out a bit. It is a mighty good guesser who can guess what skin friction will be present.

Mr. W. E. Snyder: Is there not a tendency of the caisson to go down unequally so that it is out of plumb?

Mr. George R. Johnson: Not if you have an experienced man working down there. He will take a level in with him and keep his levels on it and sink it plumb. With pneumatics a caisson out of plumb four inches would be very, very much out. With dredging, most anything is liable to happen and a foot out of plumb would be nothing unusual. You have an opportunity, when you are sinking with pneumatics, to block up on the low side and excavate on the high side and drop the caisson down. A customary way of dropping a pneumatic caisson is to do what they call blow it. They excavate down below the cutting edge perhaps a foot or so and if the shell does not follow and if there is about 30 pounds pressure, they take the men out of the caisson and open all the valves, thus taking the air off quickly, and the effect is like a blow on top with a big hammer. They immediately raise the air pressure before the water gets in. You would be surprised to see how long it takes the water to get into the caisson after the air is taken off. It is not simultaneous with the removal of the pressure, and you can still hold the water out by putting the air back quickly.

THE DESIGN AND PROGRESS OF THE FLOATING-FRAME REDUCTION GEAR

By John H. Macalpine*

DISCUSSION†

Mr. A. Peterson:‡ I have read with considerable interest Mr. Macalpine's closing remarks in connection with the discussion of his paper, and I should like to take up briefly a few of his statements so far as they refer to my discussion.

Broadly, my contention is that rigid-frame gears are running continuously, with equally high tooth pressures, at the same speeds as floating-frame gears; which shows that the complications of the floating frame are not essential to securing satisfactory operation. At the best the floating frame does *not* compensate for bending, and it cannot perfectly compensate for twist of the pinion, since the twist per unit length varies from a maximum, at the end of the pinion where the power is applied, to nothing at the free end. Moreover, the error due to bending and twist in a properly designed pinion is very small. For example, in a 6.6-inch diameter pinion, of 24-inch total face, having three pinion bearings, transmitting 2000 brake horsepower at 3600 r.p.m., the deflection, due to bending, is only 0.0005 inch and the distortion, due to twist, only 0.00046 inch as an extreme maximum, making no allowance for distribution of the pressure by varying thickness of the oil film, which from other experience is known to be a factor. If this should result in excessive pressure at any point on the tooth face, the latter will wear down until the pressure is distributed. This would happen at the maximum steady load. At lesser loads excessive pressures are not reached.

*Member, Institution of Naval Architects.

†Continued from "Proceedings," Feb. 1918, p. 71.

‡Assistant Chief Engineer, De Laval Steam Turbine Co., Trenton, N. J.

It comes to this—that, for a properly designed and constructed gear, the floating frame is not needed. Any equalizing device opens up a way for vibration and momentum effects that otherwise would not exist. The floating frame comes near to being an example of "inventing a difficulty in order to patent a remedy."

I did not mean to infer that Dr. De Laval was the inventor of the double-helical gear, but I claimed and still claim that it was due to his efforts that the modern *high-speed* helical gear was developed. He was, without doubt, the first one to operate gears at pitch speeds above 6000 feet per minute, which speed is not even nowadays exceeded to any great extent in ordinary practice.

From Mr. Macalpine's remarks the conclusion may be drawn that in the De Laval design one pinion always acts upon two gears. This is not correct. The gear reductions for the small-size De Laval turbines (up to and including 55 horsepower) built by the American company are of exactly the same general design as used for the large reductions for connection to multi-stage turbines—that is, one pinion meshing with one gear, both without and with a central pinion bearing. The English and Swedish De Laval companies, I understand, are using this construction, also, for the largest type De Laval single-stage turbines.

From Mr. Macalpine's remarks (p. 60) the impression is obtained that from a given tooth pressure and pinion diameter, I calculate the length of the helix and use such length irrespective of the ratio of length to diameter. This is not in accordance with my statements. I stated (p. 42) "The tooth pressure per inch face together with the ratio of width face and diameter of the pinion, are the principal factors governing the design of helical gears." I realize just as fully as Mr. Macalpine does that the power does not increase in proportion to the length of the helix when the ratio between length of helix and diameter of pinion is much over two, but up to this ratio the increase in power is approximately proportional to the length of the helix; which is also shown by Mr. Macalpine's curve on page 59.

In connection with Mr. Macalpine's statement (p. 62) that 691 horsepower with a tooth pressure of 325 pounds per inch

face could safely be transmitted through a pinion approximately 2½ inches in diameter at 10 500 r.p.m. by the use of a floating frame, it would be interesting to know if this is based upon experience.

In view of present public knowledge on this subject, I assume that Mr. Macalpine does not expect me to take seriously his statement that the Westinghouse Machine Company is having 100 per cent. success with its floating-frame gears. I do not believe that any other manufacturer of gears, or for that matter, anybody building machinery of similar character, could substantiate such a claim.

Mr. Macalpine's statement (pp. 64–65) to prove that tooth pressures two to three times higher than used on rigid gears *can be* and are used on floating-frame gears, is not very convincing. From the article in *International Marine Engineering,* the impression is obtained that the *Malmanger* was one of *fourteen* steamers equipped with the size gear reductions referred to and it therefore seems that, if two to three times higher tooth pressures could have been used, it would have paid to develop a special design instead of using 3400-horsepower gear reductions.

As to the three powers belonging to a ship (p. 64) I realize this, but I do not believe there is a great deal of difference in the three powers mentioned for a merchant ship. For war vessels, the difference is of course considerable.

As to gear reductions offered for municipal pumping jobs: For the one referred to in my discussion (p. 43) the specifications did *not* limit the allowable tooth pressures. In reference to the Cleveland proposition, bids for which were opened February 1914 (p. 66), it may be interesting to note that the Westinghouse Machine Company offered rigid-type gear reductions. For the 1065-horsepower turbine operating at 3600 r.p.m. the pinion diameter was 6.35 inches and the width of face 12 inches, thus giving a tooth pressure of 492 pounds per inch face; which is quite interesting in view of Mr. Macalpine's statements regarding the rigid type of gear, the tooth pressure being just as high as used on the floating-frame type. As to the De Laval gear reduction installed at Ross Pumping Station, near Pittsburgh, regarding which Mr. Macalpine has some doubts and suggests a revision

of my figures, I wish to state that this gear reduction is used as a step-up gear from 100 r.p.m. on a steam-engine to 600 r.p.m. on the pump, which is designed for a capacity of 50 000 000 gallons a day at 52 feet total head, the efficiency guaranteed being 76 per cent., requiring 600 brake horsepower, which corresponds to a tooth pressure of 955 pounds per inch face. The pump was tested after installation, and the specified conditions were obtained. I understand that the machine has been in practically continuous operation at its rated load since it was installed in 1912 and that no repairs whatsoever have been required.

Mr. John H. Macalpine: Your Society has extended to Mr. Peterson the great courtesy of making a second contribution to the discussion of my paper on "The Design and Progress of the Floating-Frame Reduction Gear," published in the February 1918 number of your Proceedings. I gladly reply to him as it will set some matters in a still clearer light.

His communication is remarkable in several respects but perhaps principally in his making no reference whatever to possible errors of alignment. It is true, he disclaimed (p. 39) ever having experienced any trouble from this cause; but the figures given on pages 61–62 seemed to indicate that the De Laval gear is very greatly hampered by misalignment and, as some readers might be influenced by these figures, I should have expected him to address himself to a more or less plausible explanation of the very light tooth pressure and small power constant used in these gears. He merely asks me if the very greatly increased pressures and power constant which I state to be safe in this case, when a floating-frame is used and nearly three-quarters of his gear discarded, is based on experience. It most assuredly is. In considering the theory of gearing, the power constant naturally presented itself to me, as I have no doubt it did to many others. This from the first has been taken as a guide in designing floating-frame gears. Large experience has proved it to be perfectly trustworthy over a large range of diameter of pinion. While the smallest pinion in use at a power constant of 4 is five inches in diameter, does Mr. Peterson mean to contend that this is a lower limit of application of this constant? If he will study the two references I gave on page 59 he will find that the truth of the

power constant rests on a very broad basis and before he can successfully controvert my claim he will have to show wherein the scientific principles involved have been wrongly applied, or what are the limits of their application.

Mr. Peterson's total omission of any reference to possible misalignment or attempt to rebut my reply to his earlier statements strongly reminds one of the reputed action of the ostrich, hiding its head in the sand to shut out from sight the danger from which it cannot escape. The vital importance of correct alignment is fully realized by his firm, which says, capitalized as below, in its advertisements, "IT IS NECESSARY THAT THE CASING BE SO DESIGNED THAT GEAR AND PINION WILL BE HELD IN ABSOLUTELY CORRECT ALIGNMENT and will be protected from outside forces and distortion. This is accomplished by the correctly designed De Laval rigid gear casing." But how the design of a casing can protect it from outside forces, either in the shops or under outside care, I cannot conceive; and the method by which a rigid casing, capable of insuring ABSOLUTELY correct alignment, can be made out of an elastic material like cast-iron, must be quite a valuable trade secret. Sir Charles A. Parsons does not profess to attain this rigidity (p. 41). Foreseeing that it was impossible and that warping or wear would be extremely detrimental, we made a gear which did not require nearly correct alignment and, as I state on page 11, I ran the trials of the experimental gear at 6000 horsepower with the gear and pinion axes diverging 1/50 inch in five feet nine inches. Fig. 4 shows that we could safely have gone much further; it also shows that this error would have been ruinous had the gears been rigid. The aligning of the pinion of a rigid gear is an extremely delicate process, as the engineers of the Westinghouse Machine Company know from large experience. Discussing the matter with Mr. A. F. Cooke, general manager of the Fawcus Machine Company, he said the beauty of our construction was the substituting of a simple process (See Fig. 11) for the tedious and excessively delicate one required in aligning the pinion of a rigid gear. Surely such a delicate adjustment is correspondingly easy to upset. Does Mr. Peterson think that in a double-reduction gear, like Fig. 19, it would be a simple matter to properly align the five shafts?

Mr. Peterson commences with a broad statement "that rigid-frame gears are running continuously, with equally high tooth pressures, at the same speeds as floating-frame gears." He has not shown that he has any running with a power constant of 4, not to speak of the 5.22 of the *Neptune*. His mode of argument is curious. He takes the case of a gear with a light tooth pressure and what for a floating-frame gear would be a very small power constant, 1.93—practically the value which I have all along contended seems to be about the upper *safe* limit for a rigid gear, though I give instances (p. 21) where this was exceeded for a short time. He then, by a slip to which I attach no importance, estimates the cross-bending at more than 11 times its proper value. He concludes that this cross-bending is of no importance for a three-bearing pinion. Taking its proper value, about 0.000044, and more than doubling it to bring the power constant up to 4—that is, raising the load to the value required to prove his contention—it certainly is of no importance. His conclusion here merely confirms what I stated on page 20. I point out, however, that for a two-bearing pinion, such as Mr. Peterson's company largely figures in its advertisements, it *is* important and will greatly limit the safe load which the pinion will carry. He also calculates the twist, which he would have found of sensible importance had he loaded his gear to a power constant of 4. He objects that this is not *perfectly* compensated by the floating frame. Surely it is better to compensate from one-half to two-thirds (p. 20) of this than, adopting the rigid gear, fail to compensate it at all. He trusts to the varying thickness of the oil film (in the bearings, I suppose) to aid in distribution of tooth pressure which, he says, "from other experience is known to be a factor." I have pointed out in *Engineering* (London), v. 102, p. 527, that careful experiment and theory teach exactly the reverse, and he is harboring a false hope. The thickness of the oil film in the bearings is infinitesimally affected by large increase of the load. Besides, can Mr. Peterson not see that a change in the bearing pressure means a change in the centers of pressure in the helixes—that is, a bad distribution of tooth pressure? Thus his argument in this respect is doubly at fault. The thickness of the oil film between the teeth, to which he may refer, is meas-

ured by a small fraction of 1/1000 of an inch and its variation could not have any sensible effect.

But if this were all, he would justly charge us with "inventing a difficulty in order to patent a remedy," for the torsional effect in a properly proportioned pinion is quite secondary. Must I point out again that annuling the effects of errors of alignment "was the object and is the most valuable function of the floating frame "(p. 20)? I have already commented on his remarkable oversight of this feature. I will return to the "vibration and momentum effects," due to the floating frame, which he seems to fear but which have never obtruded themselves in all our now extended experience.

There is one statement, however, in which he is correct, viz. that if there should be "excessive pressure at any point on the tooth face, the latter will wear down until the pressure is distributed" (p. 519). But the process is cumulative in the case of rigid gears; for successive disturbances—from wear or heating of bearings; straining of the ship, causing repeated distortions of the bed-plate such as will frequently occur in rough weather or from changes of the loading of the ship; bad adjustment of supporting wedges; screwing up loosened holding-down bolts; and what not—will shift the hard bearing of the teeth from place to place till the pinion is worn out. One must always remember that there is little elasticity in the teeth or pinion, as Mr. Peterson's calculation, especially when corrected, shows and therefore a shift of 1/1000 of an inch, even in a large gear, is an important quantity. Proper provision to guard against it could not be made by any practicable weight of bed-plate. It can be met in part by sufficiently lowering the power constant. I am not drawing on my imagination; I could cite cases of rigid gears on land—the ones I have in mind just now are not of the De Laval Company's make—where replacements had to be made every few months. On the other hand, the floating frame retains good distribution of tooth pressure in spite of a greater range of distortion that will ever be approached in practice (See Fig. 4). Its success does not depend on the realization of the hope that such disturbances will not occur—a hope which may very occasionally be fulfilled, allowing a rigid gear with a fairly high power con-

stant to succeed, but which will very many times oftener prove delusive, forcing the general use of low values.

Since this second reply was written I have received another letter from Prof. Gerald Stoney of Manchester, England, an extract from which, with some remarks, I would like to interpolate here as it throws an interesting light on the questions we are now discussing. He says:

"One important point about your design with floating frames, as compared with rigid gears, is that your tooth pressures are much higher. The usual rule here [in Great Britain] is, for pinions above about 5" diameter, to have p, the pressure in lbs. per inch run, $= K\sqrt{D}$, where D is the pitch diameter of the pinion and K is the constant. This constant is about 175 for mercantile work and up to 220 for warships; but your gears give a constant of about 250 to 280, which gives a much smaller gear. I know, of course, that you work on the rule that p varies as D, while the rule rigid gear rule is that p varies as $\sqrt{D}$ for sizes above about 5" diameter."

Since $p/D =$ constant makes the power constant, C, independent of D, obviously the rule, $p/\sqrt{D} =$ constant, will give a smaller and smaller value of C as D increases. As experience with floating-frame gears amply supports the teaching of the propositions from which the power constant is deduced, it is most interesting to inquire why a different result has been arrived at by the British rigid-gear builders.

The only reason why the rigid-gear power constant must be chosen lower as the size of the gears increases, departing from the law for similar gears, is that the gears are not made in all respects similar. In the smaller sizes, making the bed-plate relatively massive requires a much less serious addition to the weight of the gear than for a large gear. Hence the relatively thinner bed-plate of the larger gear, being more elastic, is easier to put out of alignment and the safe load is reduced. In the variation according to $\sqrt{D}$, then, we have clear proof of the great and detrimental effect of the elasticity of the bed-plate and of the allowance which always has to be made in rigid gears for possible distortions occuring at any time, a question which Mr. Peterson now completely ignores. The overall length, breadth, and depth of the gear-case (Fig. 18–19) of the largest size yet built by

The Westinghouse Machine Company, are 11 feet, 14 feet and 7 feet 6 inches, and the general thickness of metal is one inch. How many times would this thickness have to be increased to attain that great rigidity desirable in rigid gears, but which, happily, we do not require in floating-frame gears?

To sum up: The value of a solution of any problem must depend on the completeness with which the conditions are taken into account. In this problem one of the principal conditions is the possibility of misalignment from many causes which I have partially detailed. The rigid gear ignores this condition. The floating-frame gear takes full account of it and, consequently, I claim that it is on a far higher plane in its solution of the engineering problem. I trust I have now made this matter clear.

Accepting Mr. Peterson's figures for the Ross Pumping Station gear—600 horsepower, though I was assured at the Pittsburgh Bureau of Water that the power was much lower—the mystery to which I directed his attention (p. 66), and of which he has offered no explanation, is only deepened. He does not call in question the low power constants of Table IV. Why should a reduction of peripheral speed justify a rise of power constant from much under 2 to 3.48, except that the slower running gear will take longer to wear out? Or is it one of the happy exceptions to which I have referred in the last paragraph and, so far, nothing has occurred to sensibly disturb its alignment?

I wish here to call attention to Mr. Peterson's inaccurate reading and interpretation of what I wrote. On page 14 I said "Having got rid of this great delicacy [of alignment] it seemed reasonable to hope that a floating-frame gear could *safely* be loaded to two or three times the continuous *safe* load for a corresponding rigid gear." His interpretation of this is given on page 43—"As to the tremendous tooth pressures (two to three times greater than are *possible* with rigid gears) claimed to be used in connection with floating-frame gears . . . ." My language is exact, Mr. Peterson's is a travesty of it. There is obviously no reason why a rigid gear, if in good alignment, should not bear a high load and I invariably wrote with this in mind. For instance, I said on page 71, "A rigid gear can, of course, be properly aligned and, if it could be so maintained with certainty,

it could safely use a high power constant. That the power constant is reduced is proof that there is great disturbance of tooth pressure by slight changes of alignment and even with low power constants trouble is not infrequent, as I could readily prove." The Westinghouse Machine Company's engineers have frequently subjected a rigid gear to a high power constant but have never been foolhardy enough to sell one with a high power constant guaranteed. Since the De Laval Company and all other rigid-gear builders, so far as I can find, in the very large majority of cases restrict their power constants to less than 2 and we have no hesitation in using 4 with floating-frame gears, my statement is fully borne out. It is of little consequence whether my argument (pp. 64–65) is convincing to Mr. Peterson or not; what is stated there is quite subsidary to a large mass of other evidence. The facts regarding the *Malmanger* are as I state them. The contracts for the 14 ships of which she was one did not come in at one time, the orders being spread over the period from October 1915 to August 1916. The Westinghouse Company might, of course, develop a design for each special case, but surely it shows good commercial sense to reduce pattern making within practicable limits, and make one bed-plate pattern suffice for a considerable range of reduction and power. In these excessively busy times this must be done and it may be a satisfaction to some of our rigid-gear friends to be able to point to the comparatively low power constants for the lower powers and lower ratios of reduction of this range. The *upper limit* of gears now being built is 3.5 for merchant ships and 4 for naval vessels.

The three powers, even of a merchant ship (p. 64), are frequently very different indeed. Those of the *Vespasian,* which I cited, gave (1) power constant = 3.0, and (3) power constant = 1.85. Besides, it is often requested that 25 to 30 per cent. be allowed above the estimated power for the guaranteed speed, and this power may be used at any time in ships which are liberally boilered. For instance, a few days ago a ship on her maiden voyage, having a contract speed of 10.5 knots per hour, was put up for several hours to 12.2 knots. This represents a rise over the horsepower for contract speed of about 57 per cent., or a rise of about 35 per cent. in the power constant.

Mr. Peterson complains that I do not correctly represent his method of procedure in proportioning gears, and calls attention to his statements (p. 42). He quotes only one of them but not the second and much more definite one given in the first five lines of his next paragraph. The difficulty arises from the two statements being inconsistent. I carefully compared them and, relying most on the clearer one, I said (p. 60) that "Mr. Peterson *seems* first to settle . . . ." It was impossible to tell, beyond doubt, what he did mean.

He also states that my curve (p. 59) shows that until the helix length is about twice the diameter "the increase in power is approximately proportional to the length of the helix" (p. 520). I state, on page 60, that O m, Fig. 22, "is not much if any greater than two pinion diameters in a three-bearing pinion," but the curve O A shows an entirely different result from what Mr. Peterson reads from it. If P is the power and h the helix length, everyone knows that the "increase in power," as the helix lengthens, is given by dP/dh. This is a maximum at O and zero at m. To make Mr. Peterson's statement true it would have to be approximately constant.

So far as The Westinghouse Machine Company can find from its records (and the Company's engineers believe there was no exception), the tooth pressures on all the municipal pumping jobs for which it has tendered were limited by specification.

In regard to the Cleveland pumping jobs—for which bids were opened in February 1914—to meet certain circumstances of this case, The Westinghouse Machine Company did offer a rigid-bearing gear. At that time the Company had not sufficient experience to settle the power up to which it would build such gears. To-day, with more ample experience, it would not offer a rigid-bearing gear for a similar unit. But why go back to 1914, and gears with power constants differing little from unity; our practice has far passed that stage.

I have left two important questions to the end:

1. Mr. Peterson says (p. 520), regarding Dr. De Laval, "I claimed and still claim that it was due to his efforts that the modern *high-speed* helical gear was developed." What he did say (p. 39) was "that the entire credit for the introduction of the

modern high-speed helical gearing is due to Dr. De Laval." I gave proof that Dr. De Laval, whose name I greatly honor, used an invention already fully developed and in use at very considerable peripheral speeds, and that the only invention he added to it was a mistake. I consequently claim for the earlier inventors and makers a very large proportion of the credit. Accuracy of machine tools had advanced and I credit him with "probably forming the teeth with more care than had previously been done" (p. 57), but this in no way entitles him to the *entire credit.* All this refers to the end of last century. Regarding recent developments, which future historians will look on, in a very real sense, as the introduction of high-speed helical gearing, I show (p. 58) that Dr. De Laval had nothing whatever to do with them.

How can Mr. Peterson believe that any one can conclude from my remarks that "in the De Laval design one pinion always acts upon two gears"? I state (p. 3), "The cross-bending is prevented by placing the pinion, *in all but turbines of small power,* between two gear-wheels." Again (p. 57), "In the *single gear* and pinion he merely took what was then a well-known device . . . ."

Mr. Peterson seems to revel in inaccuracy.

I am much interested by his statement that he understands the English and Swedish companies are using only the single gear. This was not always so, abroad, where I can very well recollect the two large gears with a pinion between. I have no doubt I could give references had I time to examine the files of *Engineering.* Stodola, in his book on the steam turbine*, figures the two-gear arrangement. If only the single gear is now used in England and Sweden it proves, not that this is a De Laval gear except in so far as the name may be applied to gears manufactured by a De Laval company, but that the foreign builders had more engineering acumen than their friends in Trenton, N. J., and had already perceived the facts which I pointed out to Mr. Peterson on pages 61–62.

2. Mr. Peterson was quite correct in assuming that my claim of 100 per cent. success with floating-frame gears was not to be taken seriously "in view of present public knowledge on

*Aurel Stodola's "Steam Turbines." Translated from Ed. 2, enlarged, by L. C. Loewenstein. 1905. pp. 217, 219. New York: Van Nostrand.

this subject;" and, as he says, no one could substantiate such a claim. My language is obviously hyperbolic and to be understood in a purely Pickwickian sense. It may be of interest to state the material facts:

To-day there are 189 floating-frame gears in operation on land and sea. There have in all been seven cases of somewhat serious trouble. In three cases pinion teeth broke from causes unknown, as no investigation was made. Probably the troubles were due to defective material. In the *Golaa* and the *Sudbury* one of the first-reduction pinions failed at some distance from the end of the helix. Investigation showed that the cause was, beyond question, defective material. In the *Maui,* as explained on page 71, corrosion and wear were caused, especially in the turbine and gear bearings, by acid oil into which salt water had leaked causing deposition of much solid matter. The gear teeth were little affected. Lastly, the Hope Natural Gas Company has a gear actuating two gas compressors. A nut slacked off a follower bolt of one of the gas-pump pistons and caused a foul. Something had to give way. The struts of the floating frame were punched through the gear-casing. Almost at once the nut got clear, the pinion dropped back into place and the gear ran till shut down. The case was strapped where cracked and the gear has run for 18 months without further repair. The owners have paid all expenses. In no case has any structural change from the original design been made.

In a few cases, where the oil has been shut off, bearings have wiped and teeth have been scored, but this can be by no means charged to the gears. In fact only three of the foregoing seven cases are, with any degree of reason, so chargeable. But if we call the seven absolute failures, though all are running perfectly and as originally designed, from these 189 gears we have 96.3 per cent. of success. If we call them 10 per cent. failures, which I think would be greater than is reasonable, the percentage of success rises to 99.63 and I am perfectly willing to leave it at that figure.

These are the exact and *all* the material facts, to the best of my knowledge and belief. None could be produced which would sensibly change my estimate. But they in no way repre-

sent the state "of present public knowledge on this subject." There have been rumors flying around not based on fact and there has been what looks like a deliberate propaganda carried on over many years:

I called on the Pittsburgh agents of the De Laval Company and said that, for the purpose of my first reply to the discussion of my paper. I would like copies of their trade catalogues. They courteously complied and handed me Catalogue A (January 1911); Catalogue B (May 1914); Catalogue C (April 1913); and Catalogue D (March 1912). The dates given are those of the copyright.

In 1911 they appear not yet to have taken any alarm at our success. Though our experimental gear had shown great possibilities the first commercial gear, that of the Commonwealth Steel Company, was started only on March 31.

The next year, 1912, the case is different. After claiming "unique" success for their gears (Catalogue D, p. 79) they prove, by pointing (pp. 83–84) to the *Vespasian,* that such

"gears properly constructed and running in a rigid case are adaptable to the most severe service . . . and the gears are giving entire satisfaction, notwithstanding the great variations in speed, distortion of the ship's hull, pounding of the propeller and other sources of shock and vibration to which marine turbine gears are subjected."

This rather contradicts their claim of "unique" success as no De Laval Company had anything to do with the gears of the *Vespasian,* though I can find nothing in the catalogue to indicate this. Their claim of "unique" success has for many years been manifestly absurd and this catalogue should have been revised in the interests of truth.

Again, they say (Catalogue D, p. 81):

"The use of a flexible support for the pinion or the use of any flexibility in the gear or pinion itself, is detrimental, as it affords an opportunity for unequal wear and vibration. No flexible feature of any type whatever can accommodate itself to inequalities in a rapidly revolving pinion or gear on account of the short period of time in which this adjustment must be made and the mass of material involved. Furthermore, any flexible feature will permit the alignment of the pinion and gear to vary, resulting in shocks, vibration and excessive pressure on small portions of the teeth."

In the above paragraph we have the origin of Mr. Peterson's "vibration and momentum effects that otherwise would not exist" (p. 520).

I think I am not showing supersensitiveness in believing that this passage is aimed at the floating-frame gear. It shows a total misunderstanding of the functions of the floating frame, although an explanation of the device had been in print for years. No teeth can be formed quite perfectly, and no practical device can annul the effects of such errors—as I have shown, (*Engineering* [London], v. 101, p. 491, footnote). Also, I state on the same page, referring to the experimental gear:

"Although the tooth-cutting was of very high class, it was not quite so perfect as we, perhaps unreasonably, hoped. A slight vibration of the floating frame was visible, the total movement of the ends being about 1/100 in., which would give rise to inertia forces raising the mean tooth pressure per inch by a small fraction of 1 per cent. at full power. It may be that this freedom of the floating frame eased the effect of the irregularities, but it is difficult to tell."

Certainly there can be no easing of the shock if the gear is rigid, and the quiet running of floating-frame gears seems to support the belief that in this respect the device is of value. But now teeth are formed with much greater accuracy even than those of the experimental gear and the effect is infinitesimal. Apart from errors of tooth cutting, in regard to which the floating frame is an advantage, not a disadvantage as stated by Catalogue D, how can the floating frame vibrate since the pinion is in mesh with the large gear which is not only of large mass but is firmly held in its bearings? Is it intended to convey the meaning that the large gear is set in vibration also? One cannot vibrate without the other. The statements with regard to unequal wear, vibration, shocks, and excessive pressure on small portions of the teeth result from a heated imagination and want of calm engineering common-sense. Experience has not borne them out.

In 1913 (Catalogue C, p. 85), the last quotation is repeated with slight verbal changes, the principal of which is, perhaps, the insertion of the adjective "exceedingly" before "detrimental;" as the small boy whose improbable story is doubted will repeat it with emphasis adding "as sure's death," to remove reasonable doubt. Similarly for "short period of time" the 1913 catalogue

substitutes "incredibly short time." On page 87 we have "Any inaccuracy in the original cutting of the gear cannot be compensated for by flexibility in either the gear or pinion, or in their supports, or by any corrective finishing process." The first part of this I have discussed above; the second is obviously untrue. I presume the writer had learned that The Westinghouse Machine Company's practice was to run the gears under light, or sometimes full, load and, when necessary, to scrape the high spots lightly, getting an excellent bearing very quickly. A considerable proportion of the helixes requires no scraping. The De Laval Company states (Catalogue B, p. 165, 1914), that "After the gear teeth are cut they must be polished to remove *scratches* and *inequalities* left by the cutting tool. This polishing must be done by another correctly cut gear." How can a gear tooth with inequalities be described as "correctly cut"? How was the gear used for polishing made perfectly correct? Is it supposed still to be correct after it has done the polishing, or have the inequalities it has ground off had no effect on itself? The statement is hopelessly illogical. The De Laval Company grinds its gears and The Westinghouse Machine Company, if necessary, scrapes them, touching only the minute elevations requiring relief. The latter is manifestly the more intelligent process.

In Catalogue B, 1914, the De Laval engineers once more repeat (p. 165) the long quotation I have given about the evil of the flexible mounting of the pinion, again with slight verbal changes. To be sure that the statement is convincing, or to convince themselves of the truth of a statement which experience was failing to confirm, they put five lines in heavy type, commencing, "No flexible feature of any type . . . ." Further evidence of doubt is shown by two passages:

"The success or failure of certain designs and makes of gears and special devices, such as flexible supports, etc., can be determined only by years of use and constant observation. The De Laval Company from its past experience of twenty-five years of gear production would not consider the success of any device fully proved until it had been in constant successful operation for a period of from five to ten years" (p. 157).

At this time we had had floating-frame gears running most successfully for three years. They have now been running suc-

cessfully for over seven years and, consequently, it is high time a new edition of Catalogue B was issued. There is still no indication of their predictions being fulfilled but, evidence to the contrary notwithstanding, they still live in hope, for they state:

> "With improperly cut gears, it is probably true that resiliency or flexibility in the support may to a certain extent deaden noise and thus deceive the user as to the smooth running qualities of the gear; but although the shock and vibration of the gear and pinion themselves may not be transmitted to the casing and supports in such cases, the destructive action of the gear teeth is nevertheless going on and will inevitably result in shorter life, inefficiency and great impairment of reliability" (p. 165).

This seems to admit that the floating frame did soften the shock from inequalities of the teeth and that *apparently* the gear had "smooth running qualities" which, I can assure them, have completely captivated many users who, I presume, have some discrimination and are not preternaturally easy to deceive. I wish they had explained by what natural action all the trouble they speak of can be continuously taking place inside the casing and none of it to be transmitted to the outside, though the casing is somewhat like a drum. The passage is complete nonsense.

In *International Marine Engineering* for July 1916 (v. 21, p. 339) will be found an article which is very obviously inspired by the De Laval Company. On page 340 will be found a statement covering part of what I have quoted. Possibly such views are disseminated in other places which have not come under my notice.

Do the De Laval people not see that they have laid themselves open to the suspicion that they perceived the value of the floating frame which formed an advance far beyond where they were and, instead of generously recognizing it they, fearing it, originated an adverse propaganda—persistent but unsuccessful. I strongly suspect that, had they thought it so far wrong as they say, they would have allowed the natural course of events to prove its futility. No doubt, however, their efforts have gone a long way to form the "present public knowledge on this subject."

This is the psychology of the question. I will not discuss its ethics.

HEATING LIQUIDS BY ELECTRICITY

THE PAST, THE PRESENT AND THE FUTURE

H. O. Swoboda*

INTRODUCTION

The question, "Is electric water heating a success?" is in the majority of cases answered, "No, it is not and probably never will be, because electric current is entirely too expensive."

Quite a number of people support this statement with theoretical figures, pointing out clearly that the number of heat units contained in one pound of coal or in one cubic foot of gas is so very much greater than in a kilowatt-hour, that the price of the kilowatt-hour in case of competition would have to be reduced to a level so far below its cost that the central stations would suffer heavy losses and could not exist. Yet the careful observer cannot deny that the electric heating of water and other liquids has been developed during the last 15 years to such a point that the manufacture of electric water heaters, for a great many different purposes, promises to develop into a large industry which in due course of time will materially affect not only the home but all industries, as well as any institution or enterprise which is in need of hot water or steam. Indeed it is a fact that, even at this early stage of the development of the art, cases can be cited in which *electric* heating of water is the only method which will produce the desired result. To judge from the progress made during the last few years, electric water heating appears to be one of the modern methods of "doing things" which conquers its field of utility in the world in spite of its high cost, until it is finally recognized as a necessity with which present methods cannot compete at any price. It seems probable that the electric water heater will establish itself just as the gas range replaced the coal range; as the electric light superseded the gas light, and as electric cooking is about to be recognized as one of the best ways of preparing food.

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THE PAST

Development of the Art. About 35 years ago, when the first attempts were made to apply electric heat and to construct electric heaters which were suitable and desirable, numerous difficulties were naturally encountered. However, with a single exception, these difficulties consisted of nothing but the usual "baby sicknesses," and manufacturers of reputation were soon able to produce devices which could be depended upon in every way. The one difficulty just mentioned, which could not be overcome for a long time, was the lack of a suitable substance which would not be destroyed within a comparatively short time by the continued alternate heating and cooling off to which any electric heater wire —called the "resistor"—is subjected by the current. In fact, the choice of the designer of 35 years ago was practically limited to two materials for the resistor—i. e., to platinum and to the incandescent lamp.

The high cost of platinum, of course, made its application impracticable for all commercial purposes except for physicians' and dentists' instruments and perhaps for a few laboratory devices.

The incandescent lamp, on the other hand, though cheap was entirely too bulky to be successfully adopted for the great many different possibilities which presented themselves. Fig. 1, show-

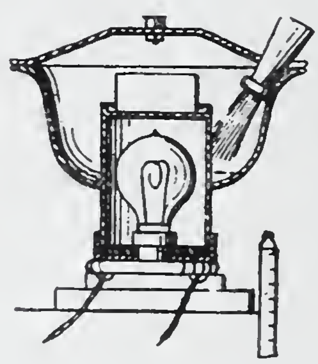

Fig. 1. Electric-Lamp Shaving-Mug.

ing an electric-lamp shaving-mug, confirms this statement, but these designs have been a matter of the past for quite a number of years. Nevertheless, a few incandescent lamp heaters are still in use and one, if not the only one, which competes successfully with more recent designs is the well-known "incandescent lamp air heater."

This condition of affairs induced the manufacturers to develop substances, mostly metallic resistors, which would withstand the strain to which resistors of electric heating appliances are subjected and which at the same time could be easily fitted to and insulated from the vessels and bodies which were to be heated. Table I shows in chronological order the various alloys

TABLE I

RESISTOR METALS AND ALLOYS

Name of substance.	Physical Condition or composition of substance.	Exclusive Mfr. or Agt.	Resistance in ohms per circ. mil. ft. at 68° F.	Temperature coefficient of resistivity for one degree F.	Specific gravity g. per cu. cm.	Approx. max. working temperature in deg. F.	Approx. melting point in deg. F.
Copper	Annealed	—	10.4	0.00393	8.89	500	1980
Platinum	Pure	—	57.4	0.00204	21.5	2700	3190
German Silver	18% Copper-Nickel-Zinc	—	200	0.000172	8.5	500	1880
German Silver	30% " " "	—	290	0.000111	8.5		2120
Ia Ia Soft	Copper-Nickel	1	283	0.00000278	8.4	700	2250
Advance	" "	2	294	nil	8.9	700	2300
Constantin	" "	3	300	0.00000278	8.6		
Ideal	" "	4	300	0.00001	8.85	700	2200
Krupp	Nickel-Steel	5	511	0.00039	8.10	1100	
Superior	" "	1	517	0.00045	8.4	1000	2400
Climax	" "	2	525	0.0004	8.14	1000	2300
Phenix	" "	4	520	0.0003	8.09	1000	2100
Excello	Nickel-Chromium	1	550	0.00009	8.9	2000	3000
Nichrome	" "	2	600	0.00024	8.15	2200	2800
Calido	" "	4	600	0.00019		2000	2800
Nichrome II	" "	2	655	0.00009	8.02	2000	2850
Calorite	" "	6	720			1000	2370
Carbon	Retort	—	4325*	not constant	1.8	6000	6700
Graphite	Acheson	—	4875*	" "	2.2	6000	6700
Carb. Filament	Treated	—	6950†	" "		3500	6700

* At 5400 degrees. F. † At 3500 degrees. F.

1—Herman Boker & Co., New York.
2—Driver-Harris Wire Co., Harrison, N. J.
3—C. Schniewindt, Neuenrade, Germany.
4—The Electrical Alloy Co., Morristown, N. J.
5—Thomas Prosser & Son, New York.
6—General Electric Co., Schenectady, N. Y.

as they were developed and applied. First, copper-nickel-zinc mixtures—known as German silver—were tried; then copper-nickel alloys, of which the "Advance" is the best known; next followed nickel-steel alloys, and finally—only 12 years ago, in 1906—the real solution of the problem was obtained when A.

M. Marsh of Detroit discovered the properties of the nickel-chromium alloys. These alloys can be operated at temperatures as high as 1750 degrees F. without being injuriously affected and their resistance is about 60 times as high as that of copper. A few other substances with similar properties have since been discovered but, so far, the nickel-chromium alloys have maintained their supremacy. In other words, *the foremost technical difficulty, which prevented the general use of electric heat, was successfully removed more than a decade ago.* This statement, however, should not be misconstrued and understood to mean that nickel-chromium is suitable for any and all purposes under all conditions. On the contrary, electric heaters have to be carefully calculated for each individual case and, if this is not done, the possibility of the "burning out" of the heaters exists the same as before. Fig. 2, for instance, illustrates one of these cases which

Fig. 2. Overheated Nickel-Chromium Wire.

is of special interest. A piece of nickel-chromium wire was overheated by the electric current to a point at which the core melted, though the outer portion remained in a solid state. In one spot, however, the solid outer crust broke and permitted the liquid core to run out, thereby changing the solid wire to a tubing. Had a flat ribbon been used in place of the round wire, the heat could

have been carried off uniformly from all portions of the area of the resistor, and in all probability it would not have "burned out."

While the development of the metallic resistors was going on, carbon, graphite and similar substances were also tried out, and to some extent with success. Since these materials, however, are at the present time applied only in special cases it would lead too far to take up this subject in connection with the topic of this evening.

THE PRESENT

Electrically Heated Vessels, Pots, Kettles and Tanks for Heating Liquids and Melting Solids with Low-Temperature Melting Points. The first electric liquid heaters which appeared on the market were the so-called "socket devices" which consume such a small amount of energy that they can be supplied from any incandescent lamp socket. They comprise coffee percolators, sealing-wax pots, sterilizers, shaving-mugs, tea-kettles, and similar portable devices which are so well known and so generally used that it will not be necessary to devote any time to them this evening. It may be stated, however, that in spite of high first cost and operating cost they have practically replaced the old-fashioned gas-, alcohol-, or oil-heated utensils wherever electricity is available.

The development through which these utensils have passed during the 35 years of their existence can best be understood by referring back to the clumsy lamp shaving-mug (Fig. 1) of the past and comparing this design with any of the modern devices, such as coffee percolators, tea-pots, or milk warmers.

Naturally the development of electric heating devices did not remain restricted to socket devices, which are limited to capacities not exceeding 660 watts. Innumerable other applications were found and their construction was very much facilitated by the use of standard heating units.

The socket devices, being self-contained, are mostly equipped with electric heaters especially designed for and permanently attached to each utensil. This method of construction of course results in the most compact form for the device, but it is not always possible to go to the expense of developing heaters of special shape. Standard heating units, on the other hand, as

disks, tubes, cylinders, flat strips, or any other form consisting of nothing but the heater itself, can be placed in vessels of any description, quite often without special provisions. At the same time it should be remembered that these heaters, if entirely immersed in the liquid, deliver as useful heat every bit of heat which they produce.

The pictures show two coil heaters (Fig. 3) and a bayonet-type heater (Fig. 4). They can all be either placed in vessels,

Fig. 3. Coil Immersion Heater.

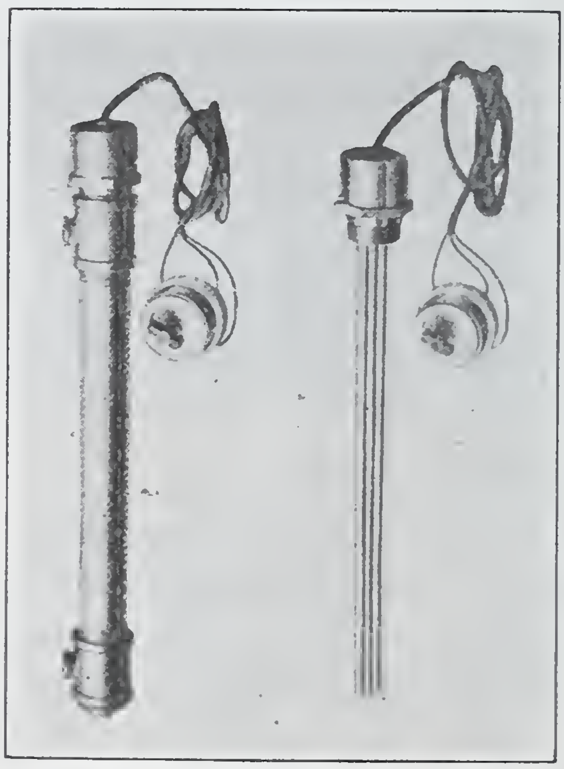

Fig. 4. Bayonet Immersion Heater.

pots, kettles or tanks without special fittings and without fastening, or they can be attached permanently, just as the requirements may be.

Standard heating units are made in almost any capacity ranging from a few watts to 10 kilowatts, and when larger capacities are required two or more units are used. The units for capacities smaller than 660 watts are usually equipped with attachment plugs for socket connection and all others, as a rule, are arranged to be connected to special electric circuits which have to be installed for the purpose.

The first large vessels for heating liquids were probably used as cooking utensils in kitchens or restaurants, hotels, hospitals and large industrial concerns for preparing soups, meats, vegetables, coffee and other foods. Although it is fully recognized that steam is in most cases more economical for this purpose, kitchens using large electrically heated vessels, pots and kettles have been found all over the world during the last 10 to 15 years. This is particularly the case where steam is expensive or unavailable. Europe—especially England, Switzerland, and Germany—has quite a number of these installations and the reports regarding them are all favorable. Fig. 5 shows the section through a stand-

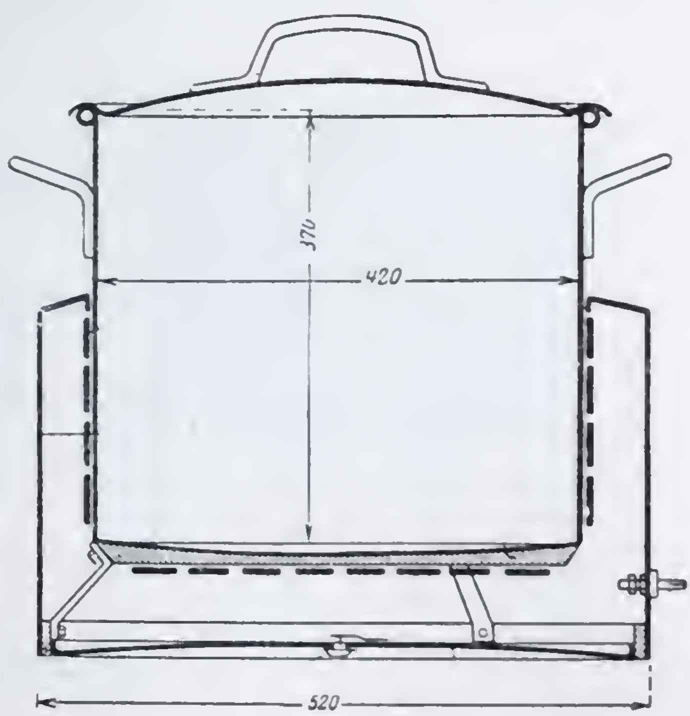

Fig. 5. Section of Large Sheet-Metal Cooking Pot.

ard type of sheet-steel cooking pot, which is made in sizes from 1½ to 50 gallons and requires, according to the size, a maximum

Fig. 6. Cast-Iron Cooking Pot.

amount of energy ranging from 600 to 6000 watts. Fig. 6 shows the exterior view of a standard cast-iron cooking pot.

The cooking utensils just described, or slightly modified designs, are used in quite a number of cases for industrial purposes —such as melting sugar, chocolate, paraffin, wax, tar, and soft metals; as well as for heating oil and various solutions.

Where a large amount of tool tempering is done, the electric oil bath (Fig. 7) is almost indispensable. Uniform temperature

Fig. 7. Oil-Tempering Bath.

control is obtained and fire hazard, uncertainty and harmful oxidation of the metals are eliminated in this process. Besides this, the work may be done successfully with unskilled labor, as

the temper is drawn by the submersion process. The tank with a flange around the top is supported by angle-iron legs, the sides are jacketed with several inches of heat-insulating material in a sheet-metal casing and the heating units, which are of the immersion type, rest in the bottom of the tank. The units are protected from mechanical injury by heavy wire mesh, which prevents free circulation of the oil through the units. Oil-tempering baths of this description can be used for temperatures up to 600 degrees F., which is just under the flash-point of most tempering oils. The tanks are made in three standard sizes ranging in capacity from 9 to 37 gallons of oil requiring a maximum input, according to the size of the tank, of from 8 to 30 kilowatts. The manufacturer claims that in the largest size tank 1600 pounds of carbon steel can be tempered every two hours.

Several complete lines of electric solder pots can also be found on the market. They are made in capacities of from 5 to 50 pounds, requiring a maximum of from 400 to 1500 watts. It is claimed that, on account of their cleanliness, the absence of fuel gases and the uniform temperature at which the metal is kept, they are superior to any other solder pots on the market. In fact, the last-named advantage results in the saving of a great deal of metal, and the danger of burns to the workman is practically eliminated.

In addition to small solder pots, large equipments for melting lead and tin alloys can be found on the market (Fig. 8). The cast-

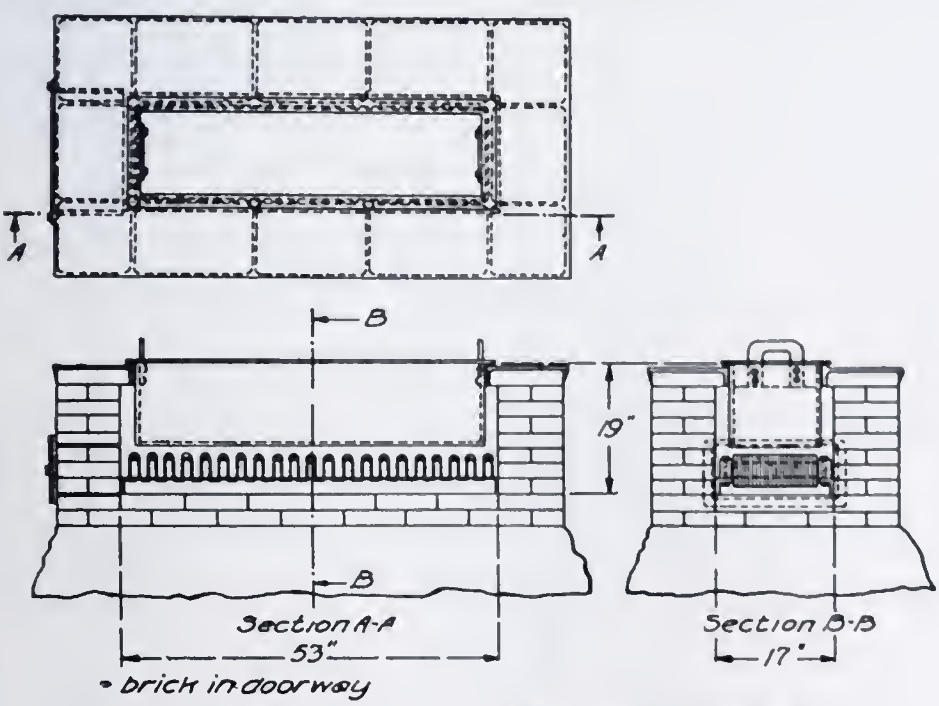

Fig. 8. Melting Pot for Lead and Tin Alloys.

iron melting pot or tank, in such cases, is supported on an iron top-plate which, in turn, rests on a foundation of heat-insulating brick. The heaters, which are of the radiant type, are mounted directly beneath the tank, but are not in direct contact with it. These pots, intended for melting soft metals, are standardized up to a capacity of 5½ cubic feet, equaling about 2400 pounds of tin, 4000 pounds of lead or 3700 pounds of babbitt. The maximum amount of electric energy required for such a tank is about 32 kilowatts, and temperatures up to about 1000 degrees F. can be applied safely.

The four illustrations just shown give an idea of the development of electric heaters as they are applied in the various industries to-day. As long as electric heat can be employed to heat chocolate or oil and to melt soft metals, it stands to reason that other materials, such as solutions, wax, tar, pitch, paraffin, type-metal and babbitt metal, can be treated in the same manner and with the same advantages; therefore it will not be necessary to present any more equipments of this nature this evening.

Hot-Water Boilers. Direct, Storage, and Circulating Heating Systems. The hot liquid which is most in demand is hot water. Numerous hot-water heaters from very small to comparatively large capacities were developed and placed on the market, but for a long time they were not favorably looked upon, and even to-day the situation is not entirely clear as to which one of the various methods of water heating may be considered the best. The amount of electric energy required for heating water is larger than for any other solid or liquid substance, and the central stations could not, and in a great many cases cannot yet, see their way clear to overcome this obstacle by offering special low rates for their energy, thereby making electric hot-water heaters a commercial success. However, the situation has changed somewhat—now that electric ranges have come into general use in the West—since people, for lack of other facilities, were obliged to heat water in ordinary cooking pots on the open burners of their ranges. As can be seen in Fig. 9, such an arrangement has an

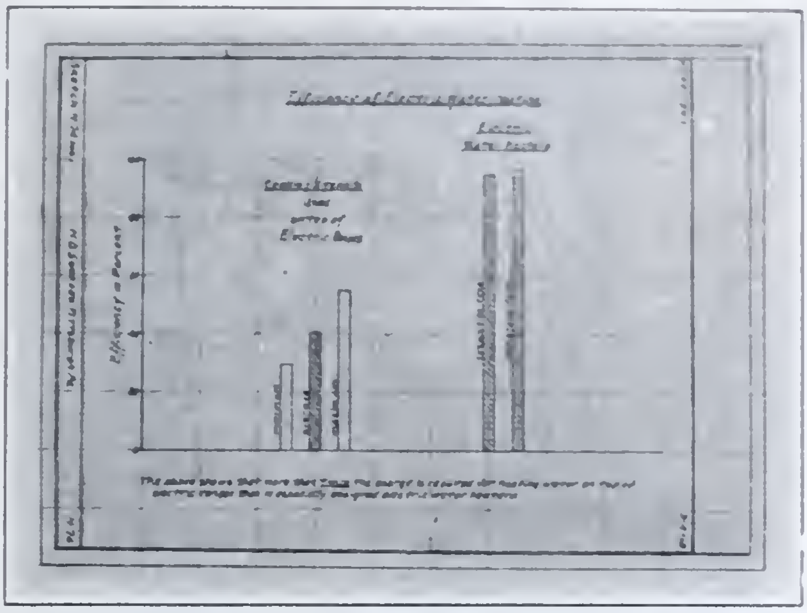

Fig. 9. Efficiency of Cooking Utensils and Water Heaters.

efficiency of only 40 per cent., whereas specially designed electric water heaters, due to an efficiency of from 90 to 97 per cent., can perform the same work considerably more cheaply. Just as soon as the users of electric ranges recognize the situation, they add hot-water heaters to their equipment, and a great many thousand are now in use in the West. In addition thereto, it seems that the central stations which experimented with low rates for water heating are satisfied that such rates are advantageous to themselves, and it is apparent that other central stations are following their example. It is, therefore, reasonable to assume that the use of electric hot-water heating will become more general with the gradual adoption of low rates for energy.

In the following I intend to give an outline of the conditions of to-day, and I will begin by showing a number of illustrations of water heaters or hot-water boilers.

Fig. 10. Hughes Small Electric Water Heater.

Fig. 10 shows a small electric water heater consuming about 750 watts. It is used to a considerable extent in connection with the electric ranges in the West.

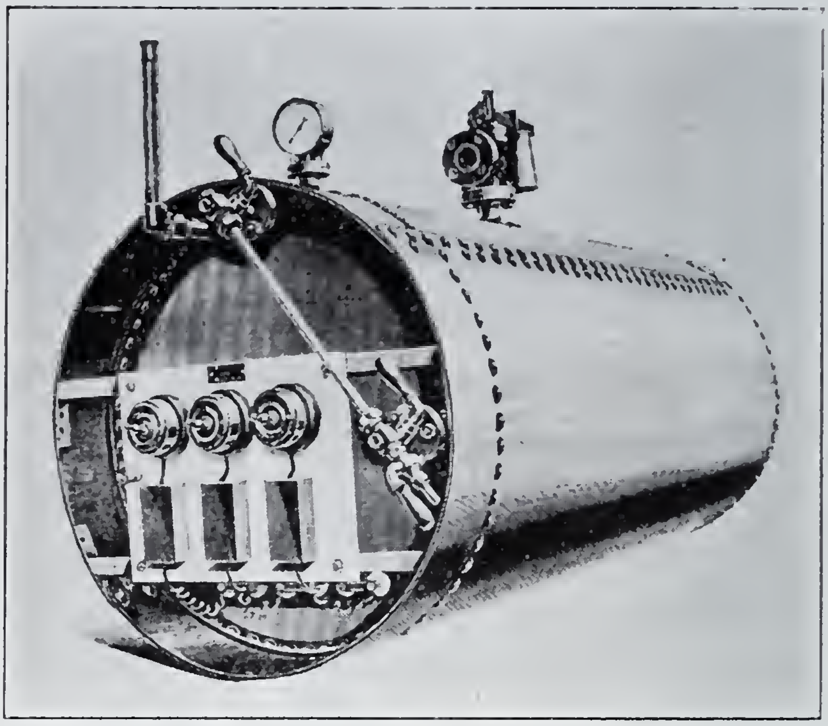

Fig. 11. Hot-Water Boilers, Hotel Moserboden.

Fig. 11 shows one of the four electric hot-water boilers at the Hotel Moserboden in Switzerland. This installation was made as early as 1904, having a total capacity of 1600 liters, with an average current consumption of 24 kilowatts. During the day, these boilers receive merely the surplus of the electric power, as they are fully charged over night. They supply all hot water for the kitchen, wash kitchen, bath, laundry, and guest rooms of the hotel, and they have been found very advantageous as well as economical. The reason of their economy is that the hotel is located high up in the mountains, developing its own water power, whereas coal is very expensive.

A similar condition exists in our own country at the Biltmore House on the Vanderbilt estate near Asheville, N. C. Since 1906 an electrically heated boiler for supplying all hot water and heat has been installed in the building (see Fig. 12). Its equipment

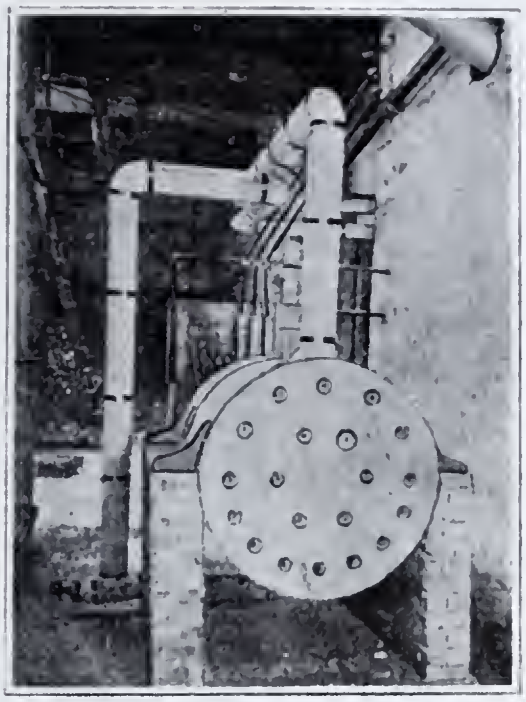

Fig. 12. Hot-Water Boiler, Biltmore House.

consists of 20 five-kilowatt, 220-volt heaters, raising 3000 gallons of water to the boiling point each day of 24 hours. Normally, however, only 6 of the 20 heaters are required to heat the necessary water. The cost of coal in this case was $11 a ton at the boilers, in 1906, whereas the price of current per kilowatt-hour was only 0.85 cents. The saving, after the change was made, amounted during the first operating year to $500 out of a total operating cost of approximately $4000 per year. This plant is still in operation.

If hot-water boilers are small, the amount of energy required to heat the water to a certain temperature within a reasonable time is not very large. However, just as soon as the quantity of water which is to be heated becomes large, either a large amount of electricity is required to obtain hot water within a short time or, if this large amount of electric energy is not available, the time of heating the water is longer than is desirable. It is, of course, not difficult to provide heaters with large energy consumption, but the central stations object very strongly to such heating devices because a number of such heaters installed on their systems exert a considerable demand on the power-house. In other words, while the consumer of the hot water is anxious to secure it within the shortest possible time, the central station desires to extend the time of raising the temperature of the water to the proper degree

of heat over as long a period as possible. These conditions have resulted, during the last 15 years, in a number of designs of water heaters which will be mentioned herewith.

At first hot-water boilers were tried with heaters of very low energy consumption, with the idea of leaving the heater in circuit all or most of the time and heating the entire amount of water gradually, whereas the hot water is withdrawn at irregular intervals in accordance with the requirements. Experiments along these lines were conducted for a number of years in this country as well as in Europe. The public, however, did not take kindly to this method of water heating and, so far as known, no such equipments are used or installed to-day.

Another attempt to reduce the large energy demand on the central station, based on the heat storage principle, was made in England, when the "Therol" water heater appeared on the market. This device employs a small electric heater to heat a cast-iron block pierced by tubes connected with a surrounding water-jacket, which in turn is fed from the water supply. A thick layer of heat-insulating material completely encloses the water-jacket. The heat developed by the electric heater gradually raises the temperature of the cast-iron block until it reaches a maximum temperature of about 500 degrees F. Whenever water is withdrawn it passes from the water-jacket through the tubes in the cast-iron block, where it is heated, to the outlet. The heat of the cast-iron block transforms the water into steam which by mixing with cold water at the outlet permits the drawing of hot water at almost any temperature. This type of hot-water heater is also a matter of the past.

The demand for hot water, however, prevented a standstill in development, and at present the circulation type of electric water heater is being used with considerable success. Its principles are identical with those of the circulation gas water heater which can be found in a great many residences and apartment houses. The appearance of the circulation electric water heater does not differ from the circulation gas water heater except that the gas heater proper has been replaced by the electric heater and that the boiler as well as the circulation pipe is carefully insulated.

The section through this circulation hot-water heater (Fig. 13) gives in detail the most advantageous relative position of the

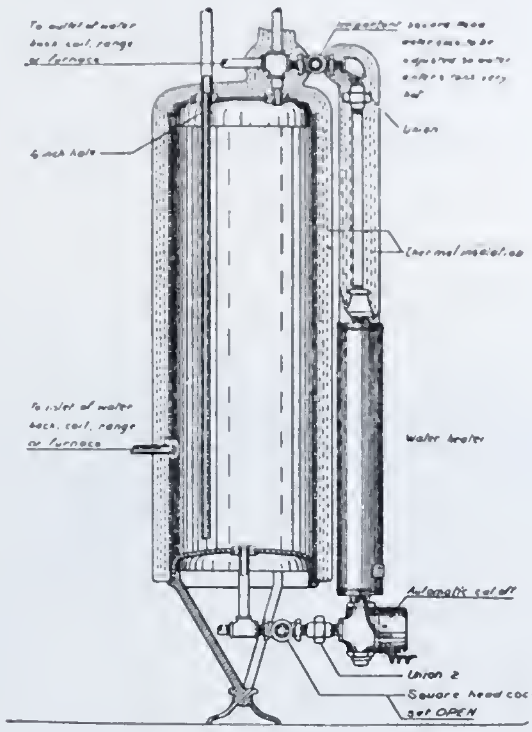

Fig. 13. Section of Circulation-Type Hot-Water Heater.

heater to the boiler, with the various connections. It also shows an automatic cut-off—so that the temperature of the water once having reached the predetermined degree of heat is maintained as nearly constant as possible—and a cock at the upper end of the circulation pipe to adjust the amount of the water flow. Fig. 14

Fig. 14. Good Housekeeping Circulation-Type Hot-Water Heater With and Without Thermostatic Control.

shows a circulation water heater of the Good Housekeeping Company, San Francisco, with and without thermostatic control.

That circulation hot-water heaters meet with success wherever they are installed can also be seen from installations for purposes other than the hot-water supply for residences. For instance, the Attix Hospital, Lewistown, Mont., has a sterilizing equipment (Fig. 15) which employs this style of heater, one for each container.

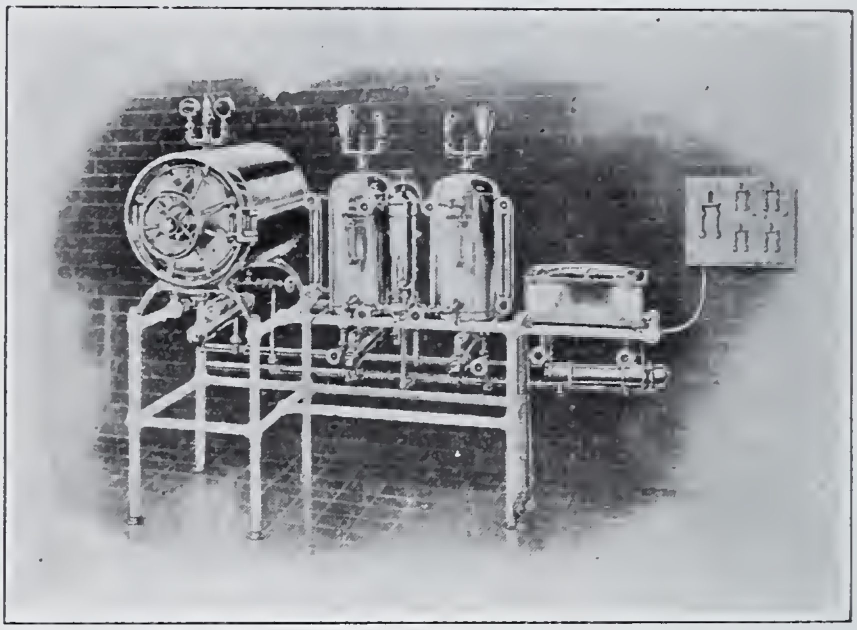

Fig. 15. Sterilizing Equipment, Attix Hospital, Lewistown, Mont.

One of the reasons why electric circulation water heaters can compete successfully with gas systems installed in private residences is that the latter, as a rule, have poor heat insulation and on this account waste a considerable amount of gas. The saving which can be accomplished by proper heat insulation can be understood by looking at the curves in Fig. 16. If a 30-gallon

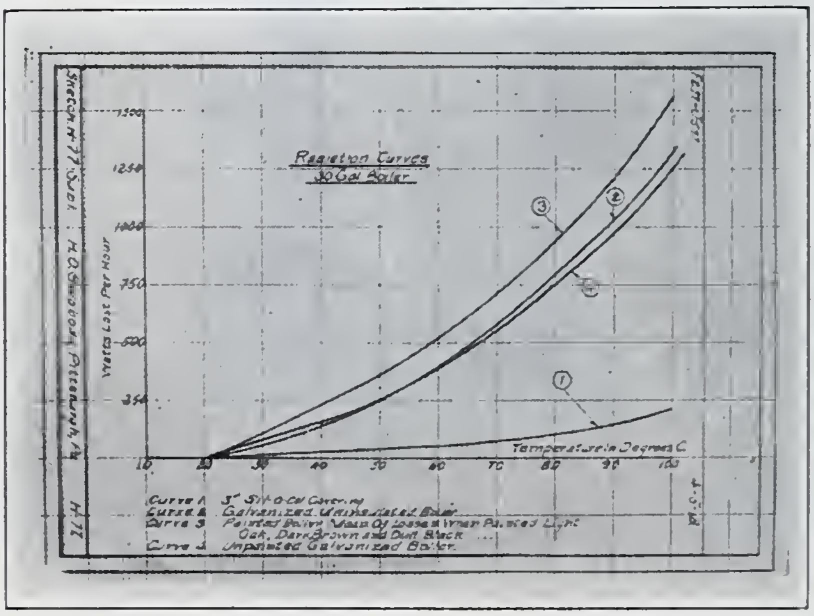

Fig. 16. Radiation Curves for 30-Gallon Kitchen Boilers.

kitchen boiler is covered with three-inch "Sil-o-cel," then not more than 250 watts per hour are lost, whereas if the boiler is left bare during the same period more than 1250 watts are lost, and if the boiler is painted this loss is liable to rise another 250 watts. Other heat insulations such as one-inch, two-inch, and three-inch hair-felt or magnesia accomplish similar results (Fig. 17). It is also

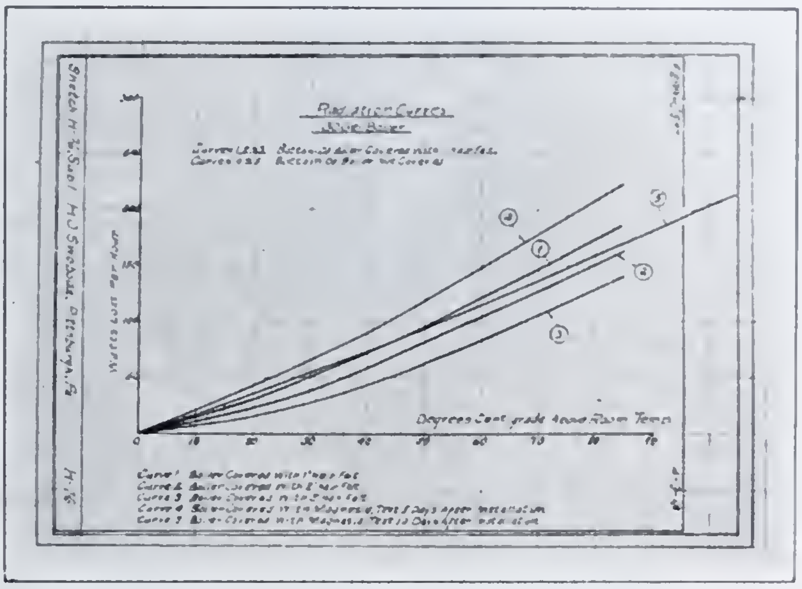

Fig. 17. Radiation Curves for 30-Gallon Kitchen Boilers.

important to cover the circulation pipe as, if bare three-quarter-inch pipe is used, more than 45 watts per foot are lost, whereas the insulating covering reduces this amount to less than one-half this figure (Fig. 18).

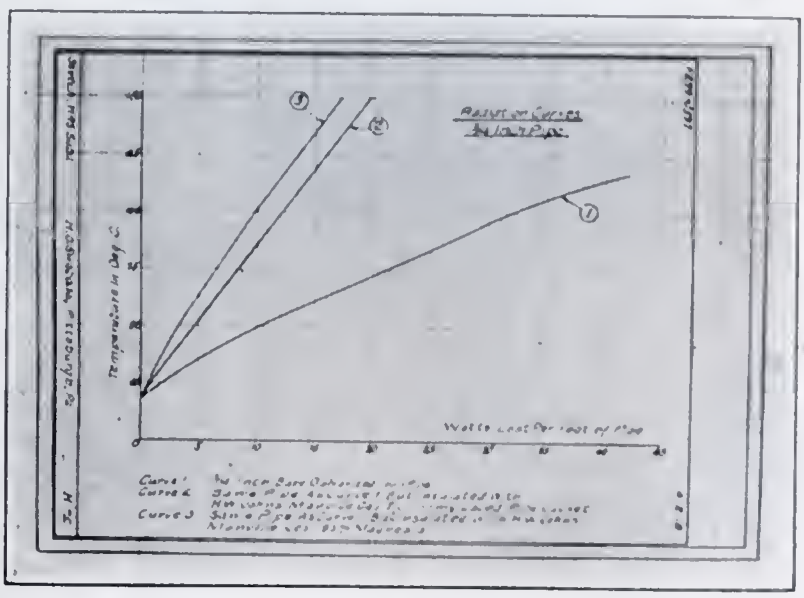

Fig. 18. Radiation Curves for ¾-Inch Pipe.

Steam Boilers. Steam can be produced by electric heat in the same manner as hot water. Engineers, considering the generation of steam by electricity, may at first glance have an idea that, in case electric energy can be cheaply produced by water-power, large electric steam-boilers will take the place of those heated by fuel. Although a few large installations are already in existence, this will not happen very often simply because when electric energy can be secured at low rates the use of steam as a carrier of energy will become very limited. The reason is that there will be no need of steam-engines or steam-turbines, because the electric motor will take their place; besides this there will be hardly any need of using the steam for heating purposes, because electric energy can be transformed into heat directly without the aid of steam. In other words, there will be only a few special cases where steam will be required as, for instance, in certain chemical processes. For such purposes only comparatively small steam-boilers will be necessary.

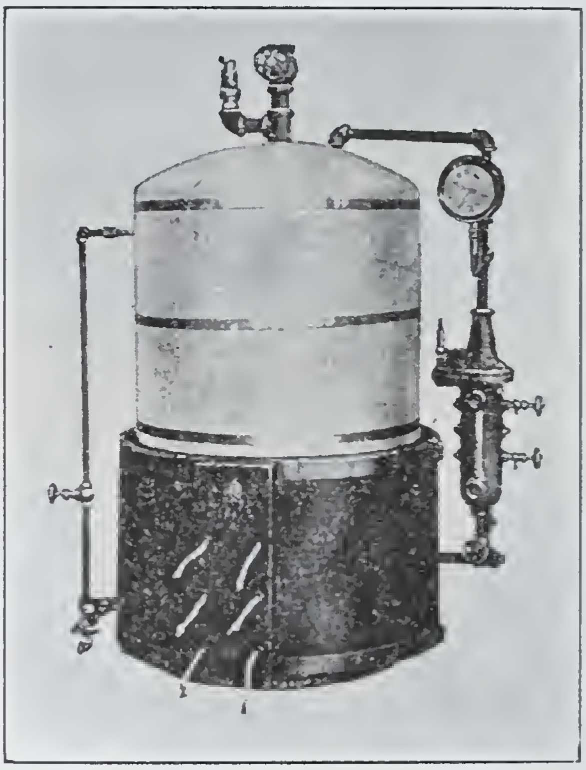

Fig. 19. G. E. Steam-Boiler.

One of the types of steam-boiler is shown in Fig. 19. It is made by the General Electric Company. These boilers are standardized up to a capacity of 200 kilowatts.

Fig. 20 shows a so-called "flash" boiler which the General Electric Company furnished for heating the trains which pass

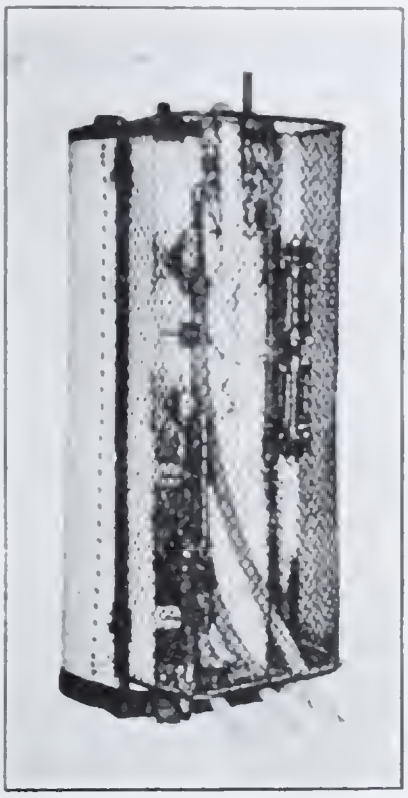

Fig. 20. G. E. Flash Boiler.

through the electric zone of the railroads entering New York City without steam locomotives. The boiler consists of approximately one-half-inch diameter steel tubes, which are heated electrically and within which the water is evaporated. At full capacity 1100 pounds of water are evaporated at a steam pressure of 100 pounds with about 400 kilowatts.

The circulation heating system described before when speaking of hot-water boilers, naturally, can be used also for generating steam. Fig. 21 shows a diagram of such an arrangement for

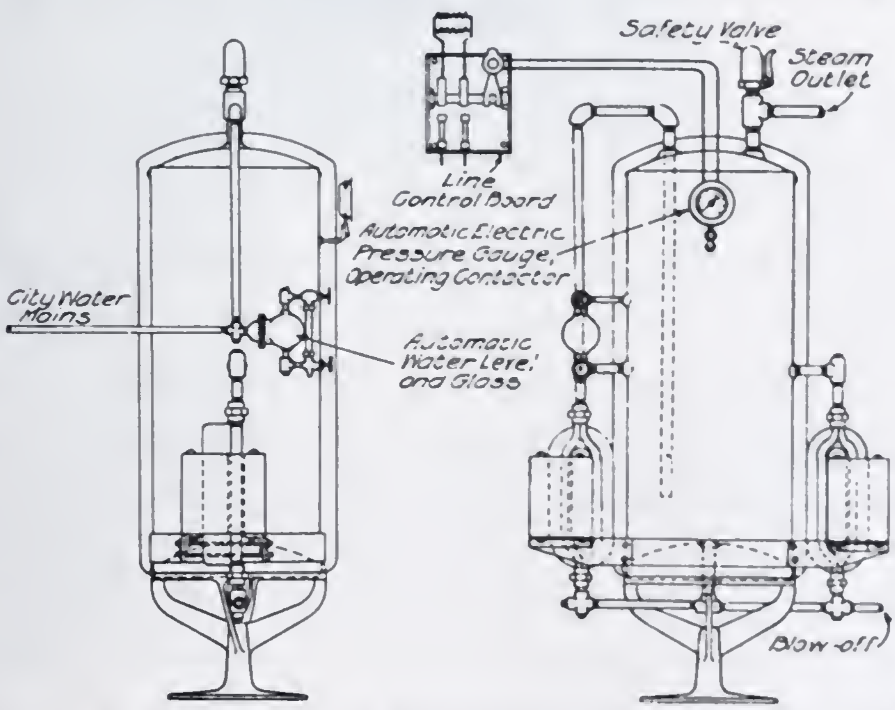

Fig. 21. G. E. Steam-Boiler With Circulation Induction Heaters.

two heaters attached to one boiler as proposed by the General Electric Company.

Another installation of an electric steam-boiler and an electric circulation hot-water heater can be found in the laundry of the Stanley Hotel, Estes Park, Colo. Electricity has been the only source of heat in this hotel for the last 10 years, and it has given satisfaction in every respect. The cost of coal delivered to the hotel 10 years ago was about $12 a ton, and this is the reason why electric energy can compete successfully.

As already pointed out, steam will be used in comparatively small quantities if electric energy can be secured at low rates. Fig. 22 shows such an application, a small electrically heated

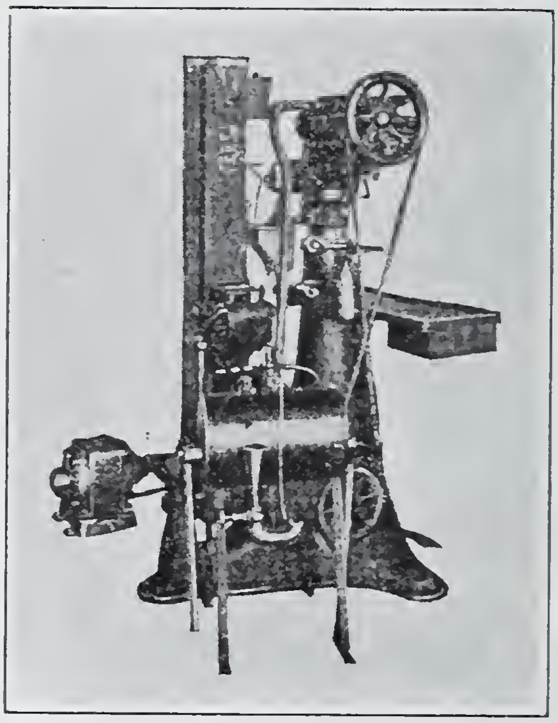

Fig. 22. G. E. Steam-Boiler Applied to Shoe-Stitching Machine.

steam-boiler which is attached to a shoe-stitching machine. While steam is available in this factory, it was found much simpler and more convenient to generate the insignificant amount of steam needed for this work in a small boiler of about 750 watts capacity directly attached to the machine, instead of running a long steam-line to the boiler room and not only suffering considerable heat losses in this line, but leakage at times as well.

The Westinghouse glue cooker is another industrial application of electrically generated steam. A bayonet heater of the standard type mounted below the water reservoir of this cooker vaporizes the water which passes through the circulation pipe into the space between the glue pot and the outer vessel, thereby keeping the glue at the desired temperature.

For the distilling of water, electric heat is also employed. One of the electric water stills is shown in Fig. 23, and a number

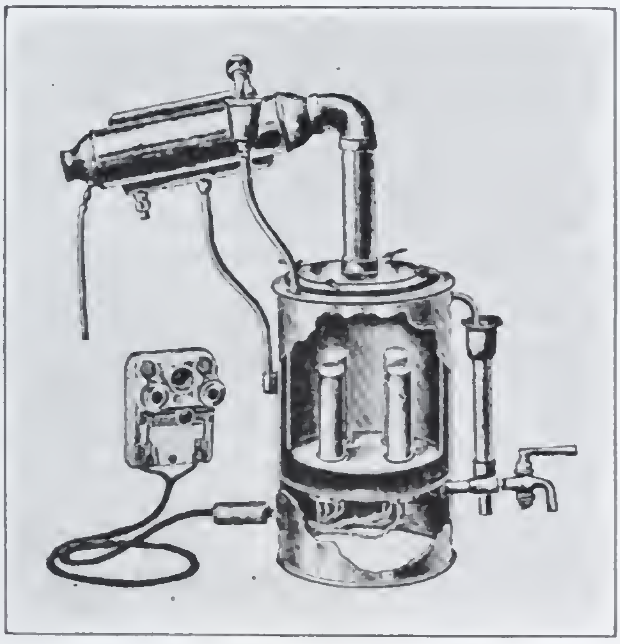

Fig. 23. Water Still.

of similar constructions are on the market at the present time. The type shown is made in capacities up to 10 gallons an hour, consuming 24 kilowatts. It is used in a great many cases for providing distilled water for storage batteries. Water stills of larger capacity are also heated by electric energy whenever the conditions are favorable.

Cost of Electric Water Heating. At the beginning of this paper I put the question, "Is electric water heating a success?" and I stated that most people reply, "No, it is not and probably never will be, because electric current is entirely too expensive." This answer is usually backed up by figures, which to some extent are presented in Table II. Comparative costs of heat generated

TABLE II

COMPARATIVE COST OF HEAT GENERATED BY COAL, GAS AND ELECTRICITY

Coal: Develops at an average a heat of 12 000 B.t.u. per lb. The efficiency of coal burning heating apparatus averages about 10 per cent. Effective heat obtained from 1 lb. of coal = 1200 B.t.u.; from 1 short ton of coal = 2 400 000 B.t.u.

Gas: Develops at an average a heat of 660 B.t.u. per cu. ft. The efficiency of gas burning heating apparatus averages about 20 per cent. Effective heat obtained from 1 cu. ft. of gas = 132 B.t.u.; from 1000 cu. ft. gas = 132 000 B.t.u.

Electricity: Develops a heat of 3413 B.t.u. per kw.-hr. The efficiency of electrically heated apparatus averages about 80 per cent. Effective heat obtained from 1 kw.-hr. = 2730 B.t.u.

Based on these figures, the same amount of useful or effective heat is generated by

1 kw.-hr. or 20 cu. ft. of gas or 2¼ lb. of coal.

TABLE OF COMPARISON

showing prices at which electricity would have to be sold, to compete with coal and gas, if there were no other advantage in using electrically generated heat.

Coal—Electricity		*Gas—Electricity*	
per ton.	per kw.-hr.	per 1000 cu. ft.	per kw.-hr.
$1.50—	0.17 cents	$0.10—	0.2 cents
2.00—	0.23 cents	0.20—	0.4 cents
2.50—	0.28 cents	0.30—	0.6 cents
3.00—	0.34 cents	0.40—	0.8 cents
3.50—	0.39 cents	0.50—	1.0 cents
4.00—	0.45 cents	0.60—	1.2 cents
4.50—	0.51 cents	0.70—	1.4 cents
5.00—	0.57 cents	0.80—	1.6 cents
5.50—	0.62 cents	0.90—	1.8 cents
6.00—	0.68 cents	1.00—	2.0 cents
		1.25—	2.5 cents
		1.50—	3.1 cents
		1.75—	3.6 cents

by coal, gas and electricity are tabulated at assumed average heat values for coal and gas and average efficiencies for the heating devices. Coal, for instance, is taken to produce 12 000 B. t. u. per pound; gas 660 B. t. u. per cubic foot; and electric energy the usual amount of 3413 B. t. u. per kilowatt-hour. The efficiency of coal-burning heating apparatus, taking small equipments into consideration, is estimated at about 10 per cent., gas about 20 per cent., and electric devices about 80 per cent. On this basis one kilowatt-hour produces the same effective heat as 20 cubic feet of gas or 2¼ pounds of coal. Applying these figures to the dollars-and-cents value of coal, gas, and electricity, it will be seen that electricity would have to be sold for less than one-fifth of a

cent to compete with $1.50 coal and 10-cent gas—that is, it would have to be sold for a little less than three-fourths of a cent to compete with $6 coal, or for 3.6 cents to equal $1.75 gas.

At the present time, electricity for heating purposes is sold to the general public at prices ranging from 1 to 10 cents per kilowatt-hour, a fair average being five cents; large consumers enjoy prices from three-fourths of a cent to five cents and, in the immediate neighborhood of a few modern hydro-electric and steam plants, large quantities of energy can be purchased as low as one-half cent per kilowatt-hour. Large mills under very favorable conditions can produce electric energy for their own use at a cost as low as one-fourth of a cent per kilowatt-hour.

All these figures indicate that, as a rule, the cost of electric energy for generating heat is higher than that of coal and gas and that, on account of this situation, other reasons than cheapness of energy must exist for the adoption of electricity. During the evening several of these reasons have been mentioned and I merely wish to repeat the most important ones herwith:

1. The possibility of fitting electric heaters into places where other heating devices either can not be placed at all or only with a great deal of difficulty.

2. The possibility of equipping each heat-consuming device with its individual heater or heaters and of locating such devices without regard to pipe systems and chimneys and without reducing the efficiency.

3. The ease with which electric wires can be efficiently and reliably installed between the source of supply and the place of consumption and practically without suffering any losses from heat radiation, leaks, and frost—losses to which all pipe-lines are subjected.

4. The entire absence of gases of combustion, dispensing with the use of chimneys and all devices for procuring the proper draft.

5. In case of coal or any other solid fuel, the entire absence of ashes and clinkers, saving thereby all labor and equipment for handling said ashes and clinkers.

6. A minimum of repairs, due to the fact that unnecessary excessive temperatures can easily be avoided.

7. Increased cleanliness and improved sanitary conditions, enabling the helpers to perform their duties better and more rapidly.

8. The possibility of producing temperatures considerably higher than can be obtained with coal, gas or any other heating method, resulting in the production of substances which it was formerly impossible to obtain on a commercial basis.

9. The possibility of controlling temperatures absolutely, not only for the degree of heat but also for uniformity over the entire heating surface, thereby reducing to a minimum the losses or waste of the material which is to be heated.

10. A minimum increase in the temperature of the surroundings, this increase being quite often not more than one-half of that unavoidable with fuel heat.

11. The saving in labor to be performed on the product, either by dispensing with it altogether or by replacing expensive skilled labor by inexperienced help.

12. The superior quality of the product, due to the uniform temperature and the absence of gases of combustion, resulting quite often in a high selling price.

13. The decrease in the possibility of accidents to workmen, due to the self-contained feature of electric heaters, resulting in a reduction of the expenses which are unavoidable in connection with accidents.

14. The reduced fire risk, resulting in the saving of fire insurance premiums.

15. The continuous decrease in the cost of electric energy, as opposed to a steady increase in the cost of all fuels.

Regarding the last-named point, it is interesting to study the dotted curve in Fig. 24. This curve indicates that if the average

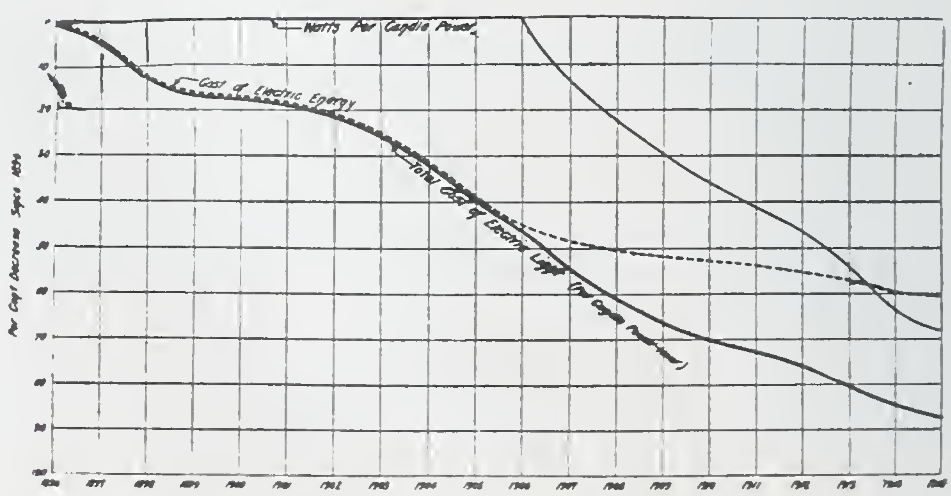

Fig. 24. Curves Showing Decrease in Cost of Electric Energy.

cost to consumers in 1890 was 100 per cent. it decreased in 1900 to about 82 per cent., in 1910 to a little less than 50 per cent., and in 1915 to about 40 per cent. Of course the abnormal conditions at present are bound to result in a temporary increase which will adjust itself in due course of time, but as coal and gas have increased more than in proportion, this temporary increase cannot retard the adoption of electric heaters.

It is impossible to express in general rules or formulas, or in dollars and cents, the savings or the advantages which may result from any one of the 15 points just enumerated; but it is certain that quite often *a single point* more than balances the high cost of electric energy and that these various points, just because their value cannot be calculated, are only too often entirely overlooked when decisions are made regarding the kind of energy which should be adopted.

As the adoption of electric heaters depends chiefly upon the cost of the electric energy, I will present to you herewith extracts from a few schedules of electric light and power companies which are in effect to-day and which more or less determine whether, in the territories covered by such schedules, electric heat can be applied to advantage. I have selected the residential or retail schedules of companies operating in prominent cities in various sections of the United States, because a considerable amount of electric hot-water heating will always be done under such schedules and because consumers of large quantities of electric energy, entitled to wholesale prices, very often have their own power-plants and do not have to depend upon the rates of the central stations in their territory. The figures which I will show, therefore, are not especially favorable for electric water heating, and in a great many cases it should be possible to secure electric energy at prices better than those named.

The Duquesne Light Company of this city established in 1914 two residential schedules (see Table III) which, with a few unimportant modifications, are in effect to-day and which are of the so-called "single rate" type. In accordance with the upper sched-

TABLE III

SINGLE RATE SCHEDULES FOR RESIDENCES

Duquesne Light Co., Pittsburgh.—Schedule A.
June 1, 1914.

Rate: 10 cents per kilowatt-hour.

Discount: 1 cent per kilowatt-hour on bills paid within 10 days from date thereof.

Minimum Guarantee: 50 cents monthly for the first 15 lamp outlets or less, and 25 cents additional for each 10 additional lamp outlets.

Lamp Renewals: Free *carbon* lamp renewals in 8, 16 or 32 candle power sizes will be furnished.

Duquesne Light Co., Pittsburgh.—Schedule B.
June 1, 1914.

Rate: A fixed service charge of 15 cents per month for each room and main hall on first floor and 5 cents per month for each room on upper floors, bathrooms and halls excepted, and 2 cents for each lamp outlet in excess of 50; and in addition 6 cents per kilowatt-hour.

Discount: 1 cent per kilowatt-hour on bills paid within 10 days from date thereof.

Minimum Guarantee: 65 cents monthly charge.

Lamp Renewals: Free *carbon* lamp renewals in 8, 16 or 32 candle power sizes will be furnished.

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ule (Schedule A) the charge in Pittsburgh for residential purposes is 10 cents per kilowatt-hour, less one cent—or nine cents net—if the bills are paid within 10 days from their date. In accordance with the lower schedule (Schedule B) the charge is six cents per kilowatt-hour, less one cent—or five cents net—if the bills are paid within 10 days from their date, plus a fixed monthly service charge of 15 cents for each room and main hall on the first floor and five cents for each room on the upper floors—bathrooms and halls excepted. The first schedule is advantageous for small consumers, whereas the second schedule is better for the large residential consumers.

As it has been found that electric cooking can compete successfully with $1 gas, if the rate per kilowatt-hour does not exceed four cents, it can readily be understood that it is useless at the present time to advocate electric cooking in Pittsburgh. As electric water heating can be made a commercial success with a rate of two cents per kilowatt-hour, it also is impossible in Pittsburgh at the present time, so long as the central station service must be employed. The result of this investigation is especially interesting, since Pittsburgh has at its disposal the best and cheapest coal and should be in a position to produce and sell electric energy at lower rates than any other section of the United States —hydro-electric plants included—had it been possible to provide the proper power equipment and distribution system. The residential rates adopted by the Duquesne Light Company do not give the large consumer a better rate than the small consumer, as the price per kilowatt-hour remains the same whether one or one hundred kilowatt-hours are consumed—in other words, they are single rates, as already stated. A great many cities, if not the majority of them, have abandoned this method of charging, in favor of combination schedules with one or more steps.

The New York Edison Company (see Table IV) charges, for instance, seven cents per kilowatt-hour for the first 1000 kilowatt-hours per month; six cents for the second block of 400 kilowatt-hours; five cents for the third block, and so on. While this rate is an improvement over the single rate of the Duquesne Light Company, the advantage is more or less imaginary for the retail consumer, because the consumption of 1000 kilowatt-hours per month is exceeded by him only in exceptional cases. Therefore New York City, like Pittsburgh, cannot at present be considered a favorable field for electric cooking and water heating.

The Philadelphia Electric Company has the same system of charging as the New York Edison Company, but it meets the requirements of the small consumer far better than does the New York company, as the first block of energy which is charged at the comparatively high rate of nine cents amounts to only 12 kilowatt-

TABLE IV

COMBINATION RATE SCHEDULES FOR RESIDENCES

The New York Edison Co., New York.—General Rate.
July 1, 1917.

Rate: 7 cents per kw.-hr. for the first 1000 kw.-hr. per month.
6 cents per kw.-hr. for the next 400 kw.-hr. per month.
5 cents per kw.-hr. for the next 800 kw.-hr. per month.
4½ cents per kw.-hr. for the next 1100 kw.-hr. per month.
4¼ cents per kw.-hr. for all over 3300 kw.-hr. per month.

Discount: None.

Minimum Guarantee: None.

Lamp Renewals: Will be furnished either at regular schedule prices or at one-half cent a kw.-hr. additional charge on the current consumed.

The Philadelphia Electric Co., Philadelphia.—Rate A Residence Service.
April 1, 1916.

Rate: 9 cents per kw.-hr. for the first 12 kw.-hr. per month.
7 cents per kw.-hr. for the next 75 kw.-hr. per month.
5 cents per kw.-hr. for all over 87 kw.-hr. per month.

Discount: None. On bills not paid within 10 days of presentation 1 cent per kw.-hr. will be added to all current billed under the second and third block.

Minimum Guarantee: 75 cents per month.

Lamp Renewals: All burned-out lamps furnished by the company will be exchanged free of charge.

The Minneapolis General Electric Co., Minneapolis.—Residential Rate.
March 1, 1916.

Rate: 8½ cents per kw.-hr. for the first 3 kw.-hr. per room per month.
6 cents per kw.-hr. for the next 3 kw.-hr. per room per month.
2½ cents per kw.-hr. for all additional current.

Discount: 5 per cent. for payment within discount period.

Minimum Guarantee: $1.00 monthly.

Lamp Renewals: Nothing stated.

Puget Sound Traction, Light & Power Co., Seattle.—Residence Rate.
April 1, 1915.

Rate: 5½ cents per kw.-hr. for the first 45 kw.-hr. per month.
2 cents per kw.-hr. for all over 45 kw.-hr. per month.

Discount: None.

Minimum Guarantee: 50 cents monthly.

Lamp Renewals: All burned-out *carbon* lamps furnished by the company will be exchanged free of charge.

hours, and the second block to 75 kilowatt-hours, and any excess of energy used over and above the 87 kilowatt-hours is charged at five cents.

The system of charging adopted by the Minneapolis General Electric Company is similar to the practice adopted in New York and Philadelphia except that the amounts of the first and second blocks vary with the number of rooms which a residence or building may have. It should be noted, however, that the first two blocks are very small, and that any excess is charged at a rate of 2½ cents.

The Puget Sound Traction, Light & Power Company, of Seattle, has adopted the same system of charging as New York and Philadelphia, but offers the consumer considerably lower rates, as the highest charge is 5½ cents for the first block of not more than 45 kilowatt-hours and only two cents for all energy consumption in excess of this first block.

It can be seen from the above schedules that it is hardly advisable to use electric cooking in New York and Philadelphia, but that electric cooking is well able to compete with gas cooking in Minneapolis and Seattle.

The schedules so far shown are made by the electric companies for the sale of energy for all purposes, no matter whether the current is used for electric light, household appliances—such as sad-irons and fan motors—or heating appliances. Other electric companies have considered it advisable to establish special rates for heating appliances, especially for cooking, and lately also for electric water heating (see Table V).

The Edison Electric Illuminating Company, of Boston, for instance, has adopted a schedule whereby 10 cents per kilowatt-hour is charged for the first 10 kilowatt-hours consumed, and only two cents for all excess. Such a rate enables the consumer to use electric ranges as well as electric water heaters in competition with gas- or coal-heated ranges.

TABLE V

SPECIAL RATE SCHEDULE FOR COOKING AND HEATING

The Edison Electric Illuminating Co., Boston—Miscellaneous Energy Rates.
January 1, 1917.

Rate: 10 cents per kw.-hr. for the first 10 kw.-hr. per month.
2 cents per kw.-hr. for all additional current.

Discount: Not stated.

Minimum Guarantee: $9.00 per year.

SPECIAL RATE SCHEDULE FOR COOKING AND HEATING

Potomac Electric Power Co., Washington.—Schedule H.
August 1, 1916.

Rate: 10 cents per kw.-hr. for the first 10 kw.-hr. per month.
3 cents per kw.-hr. for all additional current.

Discount: None. On bills not paid within 10 days of presentation an amount equal to 10 per cent. of the bill is added.

Minimum Guarantee: $1.00 monthly.

SPECIAL RATE SCHEDULE FOR COOKING, HEATING AND POWER

Pacific Gas & Electric Co., San Francisco.—Schedule No. 152.
April 1, 1914.

Rate per kw.-hr.:
4 cents for the first 30 kw.-hr. per kw. of active con. load.
2 cents for the next 90 kw.-hr. per kw. of active con. load.
1½ cents for all over 120 kw.-hr. per kw. of active con. load.

Discount: None.

Minimum Guarantee: Up to 5 kw. con. load, $5.00; each additional kw., $1.00.

SPECIAL RATE SCHEDULE FOR WATER HEATING

Great Western Power Co., San Francisco.—Schedule No. 305.
September 1, 1917.

		Per month	
Rate:	1500-watt heaters in connection with range,	$2.50;	without, $ 4.00.
	3000-watt heaters in connection with range,	5.00;	without, 8.00.
	5000-watt heaters in connection with range,	8.50;	without, 13.50.

Water heaters must be approved by the company.

Discount: None.

Minimum Guarantee: Same as flat rate provided.

The Potomac Electric Power Company, of Washington, adopted a similar schedule, charging 10 cents for the first 10 kilowatt-hours per month and three cents for all excess.

The Pacific Gas & Electric Company, of San Francisco, charges four cents per kilowatt-hour for the first 30 kilowatt-hours, steps downs to two cents for the next 90 kilowatt-hours, and to 1½ cents for all excess.

The Great Western Power Company, of San Francisco, has found it advisable to make a special rate for electric water heaters, charging for water heaters of 1500 watts capacity $2.50 per month flat if used in connection with an electric range, and $4 per month flat if no range is used. For larger capacity water heaters the flat-rate charges are correspondingly higher.

Before closing the subject of rates, I desire to present a few figures showing the cost of water heating in Pittsburgh and in San Francisco (see Table VI). In accordance with this calculation,

TABLE VI

COMPARATIVE COST OF HEATING WATER BY ELECTRICITY AND GAS

1 gallon of water weighs 8.3 lb.
30 gallons of water weighs $8.3 \times 30 = 249$ lb.
1 gallon of water requires $1 \times 8.3 = 8.3$ B. t. u. for raising the temperature 1 degree Fahrenheit.
30 gallons of water require $8.3 \times 30 \times 50 = 12\,450$ B. t. u. to raise the temperature 50 degrees Fahrenheit.

Electricity	*Gas*
1 kilowatt-hour equals 3412 B. t. u.	1 cu. ft. of gas averages 660 B. t. u.
12 450 B. t. u. = 3.65 kw.-hr.	12 450 B. t. u. = 18.9 cu. ft.
If heater has an efficiency of 95 per cent., it will require 3.65/0.95 = 3.84 kw.-hr.	If heater has an efficiency of 60 per cent., it will require 18.9/0.60 = 31.5 cu. ft.
If electricity can be purchased at 1 cent per kilowatt-hour, it will cost $3.84 \times 1 =$ *3.84 cents* to raise the temperature of 30 gallons of water 50 degrees.	If gas can be purchased at $1.00 per 1000 cubic feet, it will cost $31.5 \times 1.00/1000 =$ *3.15 cents* to raise the temperature of 30 gallons of water 50 degrees.

In *Pittsburgh*, with a price of 6½ cents per kilowatt-hour, the cost of electric heating would be 25 cents.

The cost of gas heating with 1000 B. t. u. gas at 30 cents per 1000 cubic feet would be 0.62 cents.

In *San Francisco*, with a price of 1½ cents per kilowatt-hour, the cost of electric heating would be 5.76 cents.

The cost of gas heating with 660 B. t. u. gas at 75 cents per 1000 cubic feet would be 2.36 cents.

it costs 25 cents in Pittsburgh to heat one gallon of water electrically, and only 0.63 cents if natural gas is applied. The costs in San Francisco are 5.76 cents and 2.36 cents, respectively. In other words, heating water by electricity in Pittsburgh costs about 40 times as much as heating by gas, whereas in San Francisco the cost is only about twice as much. The result is that Pittsburgh has no electric water heaters whatsoever in use, whereas several thousand are in successful operation in San Francisco, and their number is on a continuous increase.

In studying the schedules adopted by the electric light and power companies in the United States and Canada, it will be found that west of a line drawn from Chicago to New Orleans—as well as around Niagara Falls and in Canada—the rates are sufficiently low to warrant the adoption of electric heaters not only for cooking but for a great many industrial appliances, and eventually for water heating. Furthermore, it can be noticed that the number of central stations considering electric heating loads desirable is continuously increasing, as new low rates favorable for electric heating are being established all the time. Therefore it seems quite reasonable to assume that the adoption of electric heating in place of fuel heating, in the present state of the development of the art, increases just as fast as the charges for electric energy decrease. Of course, the abnormal conditions at the present time may retard the continuous decrease in the cost of electric energy, but there is no doubt that, with the re-establishment of peace, the rates will continue to decrease, and thus widen the field for electric heaters.

THE FUTURE

Instantaneous Water Heaters. The devices so far described have all been tried out carefully and all are manufactured more or less in large quantities. The electric companies have made thorough and painstaking tests of the various equipments, and, in general, the devices are well known to the public. The development of the electric heating line has therefore reached a point which permits a steady production by the manufacturer to meet the continuously increasing demand from the public. Nevertheless, matters have not as yet reached a point where the first stage

of the development can be considered complete. So far as water heating in particular is concerned, considerable changes from the present practice may be expected sooner or later which will enlarge the field of electric heating with a great deal more rapidity than at present. The development to which I have special reference is that of instantaneous water heaters.

I believe the small, portable, instantaneous water heater found on the counters of drug stores and candy stores is pretty well known. This device furnishes a glass or cup of hot water instantaneously and by its ever readiness it has replaced other hot-water heating devices for the same purpose. On the same principles, instantaneous water heaters for running water can also be found on the market. The faucets of these heaters have three distinct positions, one for the shut-off position, another one for supplying hot water, and a third one for supplying cold water. Heaters of this kind can be installed without a great deal of difficulty in any place where hot water is required—for instance, on wash-basins or in connection with bath-tubs. The instantaneous water heater shown in Fig. 25 is arranged in such a manner

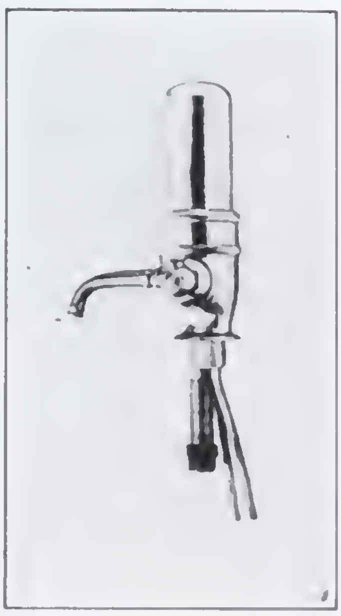

Fig. 25. United Sales Instantaneous Water Heater.

that it can be mounted in place of any ordinary water faucet at wash-basins, and its extreme simplicity has given this device a ready market out West, wherever the cost of electric energy is low. The entire heater (Fig. 26) consists of nothing but one

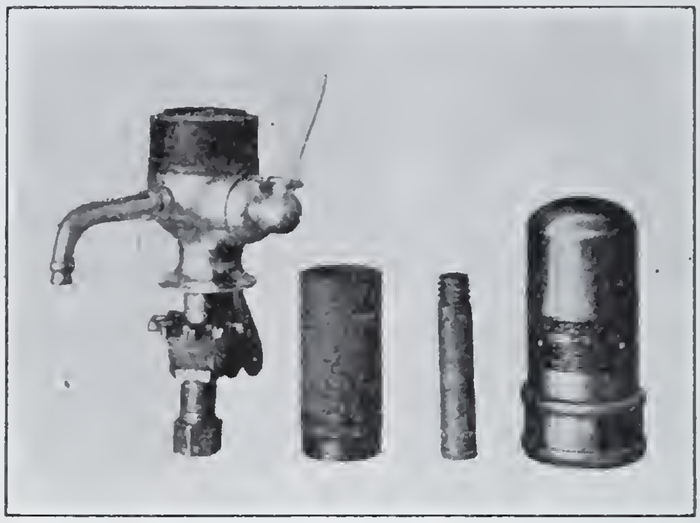

Fig. 26. United Sales Instantaneous Water Heater, Exploded View.

graphite electrode and one carbon cylinder, which are mounted on top of the faucet and covered up by a porcelain and metal cap. Whenever the deposit of dirt which is left by the running water becomes too thick and is liable to prevent the passage of the water, anyone can take the heater apart and clean and reassemble it, without special experience and without a great deal of skill.

The general adoption of such instantaneous hot-water faucets would mean a considerable advance in the art, as hot-water pipe systems and hot-water boilers in private residences, as well as in all other buildings, could be abandoned and these instantaneous hot-water faucets attached to the cold-water pipe system instead. Not only would the first cost of the equipment be materially reduced, but also the expense of operation, if one considers that at present, whenever hot water is required somewhere in a building, all the water standing in the pipe between the boiler and the place of consumption has to be heated in addition to the amount of hot water actually required. The instantaneous hot-water faucets eliminate this waste, as they produce the heat at the place of consumption, thereby saving both water and heat. This, again, means that the high price of electric current is offset to a considerable extent by the smaller amount of heat which has to be produced.

The reason why these instantaneous water heaters have not so far been adopted for general use is that they require a considerable amount of electric energy for a few seconds or minutes

and, at present, the central station power-houses have not sufficient capacity to meet these large and short demands. It is for this reason that the leading manufacturers of electric heating appliances have not not as yet taken up the production of instantaneous water heaters in large quantities. It seems, however, that the increasing capacities of the power-houses and the demand made by the public for such devices will gradually bring the instantaneous hot-water heater into the foreground and to a considerable extent open up the field of preparing hot-water by electricity. In fact, the first step for introducing these instantaneous water heaters in large quantities has been taken in the West, where they can be found installed on the electric systems of all large central stations; and as soon as the advantages are recognized and understood by the consumers, there is no doubt of the gradual adoption of these heaters.

I hope that the talk of this evening will give you a fair idea of the art of electric heating, its place in the house and in industry and its prospects for more extensive application.

In conclusion, I wish merely to repeat that electric heating does not offer any serious obstacles to the engineer, and a more general use is bound to come with the continuously decreasing cost of electric energy; the increase in the capacity of the electric power-plants; and a more intimate knowledge of the various advantages of electric heaters, so that the designing engineer can plan his equipments to the best advantage and not merely replace fire grates, gas-burners or steam-pipes by electric coils.

DISCUSSION

Mr. W. E. Snyder, *President*:* Gentlemen, I think we have had a very splendid paper. I regret that more members of the Society could not hear it. I want to say, as far as I am concerned —and I suppose the same thing is true of a good many of you-- that while I have had to do for a good many years with the operation of boilers, and making steam, and the production of electric current, I know practically nothing about this subject. It is a revelation to me.

I just wondered, while this paper was being read, if a condition could develop—where water heaters were being used about houses or for various miscellaneous purposes—by which an electric shock might be received by any one; or how is the current actually used and how is the voltage reduced at the heater?

In the paper presented, the comparison of cost of power to concerns in this city is very interesting to me. I have made some comparisons of this kind myself with greater quantities of power, and there seems to be no logical reason for such differences. Mr. Swoboda has pointed out that the cost of power for heating water in Boston is less than in Pittsburgh, and I have noticed a similar condition with larger quantities of power. Also, very often the terms under which power is purchased are so complicated that it is not possible for any one to foretell what the power cost is going to be. The total cost is composed of two or three parts—one part a service charge, one part depending upon the consumption, etc. So conditions around here, at least up to the present time, have been very unfavorable to the use of electric power in large quantities for such purposes as mentioned in the paper.

Is there any discussion, gentlemen?

Mr. Samuel E. Duff:† I don't know, Mr. President, that I can contribute any more to the details than you can. I stand almost in the same position—having for a number of years used

*Mechanical Engineer, American Steel & Wire Co., Pittsburgh.

†Consulting Engineer, Pittsburgh.

things that have something to do with electricity, but not being sufficiently versed in the subject of the evening to properly discuss it. Like you, I am very much astonished at the degree to which the work of manufacturing electric heating devices has been extended. We are all familiar with the electric curling-iron and the coffee percolator and the cigar-lighter, etc., but the number of uses indicated by the speaker is very much larger than I realized.

The point brought out by the President, in regard to shock to any person using this apparatus, seems to me to be important. Possibly in one way it is not, because unless apparatus of this kind is safe it will not be used. Personally, I think I would be a little bit dubious about an instantaneous water heater, from the experience I have had with cigar-lighters and things of that kind. They seem to have a penchant for short-circuiting when you do not expect it. It seems to me I would look around very carefully before taking a bath in a bath-tub if I saw electric wires running into the water faucet.

Another thing that impressed me in connection with the paper was the cost of electric power in Pittsburgh. The question of the use of electric power for manufacturing establishments is something that I have had a good deal to do with, simply from the commercial or financial standpoint. The trouble I have found is the unreliability of the service. I have in mind a manufacturing establishment near Pittsburgh which has a business of about $5000 a day when in full operation. At any time, even under the worst conditions, it has a production value of $2000 a day. That manufacturing establishment is supplied with electric power from a commercial station. The electric power costs only about $300 to $500 a month, so that the cost of power is negligible compared with the value of the operation of the business. An interruption of that power surely means a financial loss, but interruptions of that power seem to be caused by things which would be, apparently, very easily corrected if the electric power producing companies would take the same pains with their business that other people are forced to take with theirs. A storm comes along and blows down some of their lines, or perhaps circuit-breakers fly out or something else equally preventable happens, and there seems to

be nobody around to correct it for hours; or to say when power will be supplied.

It seems to me, in connection with this very fine paper showing the wonderful possibilities of the application of electric energy to all kinds of uses, that we ought to keep in mind the fact that public utility companies which undertake to furnish us with this energy must devise some means of giving a constant and unfailing source of energy if they want to build up an asset of satisfied customers. They do not, apparently, in this neighborhood at least, pay enough attention to taking care of the consumer by supplying a constant, unfailing source of electric energy. The consumer cannot instantly change to coal or gas energy, so when the electricity fails he is out of pocket and out of humor.

Author's Closure: There is no danger of receiving a shock from the water passing through electric water heaters, as in most cases the electric current is thoroughly insulated from the water. Some designs, however, do not insulate the current from the water, but ground the point in which the current makes contact with the water, obtaining thereby the same result—i. e., that no shock can be obtained. The danger of receiving a shock from the water of an electrically heated water heater is just as little as that of receiving a shock by touching the frame of a dynamo or motor.

It is hoped that the paper has shown that electric water heating is feasible and that the adoption of this method will become more general just as soon as the capacity of the power-houses increases and the cost of electrical energy decreases.

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NEW AND LITTLE-KNOWN METHODS OF CALCULATION OF GIRDERS, BEAMS AND ARCHES

James S. Martin*

It is with hesitation that we use the word "new" in connection with the methods described in this paper, for the reason that whenever some one has developed a system of calculation which he believes to be new, he is very apt to find to his sorrow that some one has preceded him. In the preparation of this paper we have been forcibly reminded of the fact that there is no new thing under the sun.

The wise man of old also reminds us that "of making many books there is no end" and, as this saying is true in the line of engineering literature as well as in any other line, it is next to impossible for a busy man to keep in touch with all the books which are issued from time to time on the various phases of engineering. In fact, during the preparation of this paper the writer, in searching through a number of works on structural engineering, has found some of the methods herein described, developed by other writers. At first the writer was inclined to omit the mention of these methods, but since the line of development followed here is different from the lines followed by any other author of whom the writer knows and, as we expect to carry the development farther than has been done by any other author, the writer decided to present the methods, giving credit to those who have preceded him whenever it is possible for him to do so.

The question has been asked, "Why did you take such a varied subject?" J. A. L. Waddell in the preface to his book "De Pontibus" gives as his reason for using a Latin title, that it was the first opportunity he had had in 22 years to use his knowledge of that language and he was afraid he would not have another for 22 years more. For similar reasons the writer asks

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the privilege of presenting several phases of the subject. Some of these methods and formulas have been used by him for 15 or more years and this is his first opportunity to present them, and if he fails to do it now, it may be another 15 years before another opportunity presents itself and in the mean time some other party might step in before him.

We will discuss first the determination of the economical depth of plate-girders.

Many writers have approached this subject but none, to the writer's knowledge, has thoroughly discussed it. By some it is held useless to try to develop a mathematical formula which would be practical. Others claim that experience will produce better results than mathematical formulas. The formulas which we present to-night are tested by years of trial, and the writer will testify to the fact that they are practical; that they enable a person to determine the most economical form of girder in a short time; and that results are obtained as rapidly as by any other method of calculation.

Some years ago, the writer gave some of these formulas to a structural-steel designer for one of the largest construction companies in the country. A few days later this designer told the writer that he had just designed some girders before the previous conversation, and that after going back to the office, he had decided to try these formulas with the result that he had cut four or five pounds to the foot out of the weight of these girders. At another time a designer for another large company tried to demonstrate the superiority of the older methods, but failed.

Of course it must be remembered that any formula must be applied with judgment. Nothing will take the place of common-sense in any line of work. It may be that the reason the use of such formulas for determining the economical depth of plate-girders has not found more favor with the engineering profession, lies in the fact that a single formula has been used indiscriminately. Different formulas must be used for different circumstances, yet the number of these formulas which it is necessary for one to keep in his mind for practical work is really very limited. The formulas which we present here to-night will cover nearly all circumstances which are likely to occur in common

experience. While we present 48 formulas, yet in practical work the writer has had occasion to use only a very small part of them except in rare instances, and has found that those which he ordinarily uses are as easy to keep in mind as the ordinary formulas for determining the bending moments of beams. We have divided girders into four classes, and into three divisions in each class. The classification is explained in the notation for the tables presented. With regard to the divisions, you will note that the third division practically corresponds to the method of taking the section modulus of the entire section.

When the writer was introduced to the mysteries of designing girders, he was told that one-twelfth of the length was generally the most economical depth. You will notice that the length has very little to do with the economical depth. Just recently the writer has had occasion to design some girders in which the economical depth was more than one-fifth of the length.

TABLE I*

SIMPLE FORMS, NO ALLOWANCE BEING MADE FOR STIFFENERS AND FILLERS

Class	Division	y	a	a_m	A	A_a
1	I	$\sqrt{\frac{2M}{ft}}$	$\frac{yt}{2}$		$2yt$	
1	II	$\sqrt{\frac{8M}{3ft}}$	$\frac{yt}{4}$		$\frac{3yt}{2}$	
1	III	$\sqrt{\frac{3M}{ft}}$	$\frac{yt}{6}$		$\frac{4yt}{3}$	
2	I	$\sqrt{\frac{13M}{8ft}}$		$\frac{8yt}{13}$		$2yt$
2	II	$\sqrt{\frac{104M}{51ft}}$		$\frac{19yt}{52}$		$\frac{51yt}{32}$
2	III	$\sqrt{\frac{78M}{35ft}}$		$\frac{11yt}{39}$		$\frac{35yt}{24}$
3	I	$\sqrt{\frac{211M}{128ft}}$		$\frac{128yt}{211}$		$2yt$
3	II	$\sqrt{\frac{1688M}{813ft}}$		$\frac{301yt}{844}$		$\frac{813yt}{512}$
3	III	$\sqrt{\frac{1266M}{557ft}}$		$\frac{173yt}{633}$		$\frac{365yt}{192}$
4	I	$\sqrt{\frac{7M}{4ft}}$		$\frac{4yt}{7}$		$2yt$
4	II	$\sqrt{\frac{56M}{25ft}}$		$\frac{9yt}{28}$		$\frac{25yt}{16}$
4	III	$\sqrt{\frac{42M}{17}}$		$\frac{5yt}{21}$		$\frac{17yt}{12}$

For comparison of divisions in class 1 $A_{(I)} : A_{(II)} : A_{(III)} :: \sqrt{4} : \sqrt{3} : \sqrt{\frac{8}{3}}$
$:: 1 : 0.86603 : 0.81649.$

Table I gives the simple forms in which we make no allowance for stiffeners and fillers. The formulas in Class 1 are practically all that one needs to keep in mind. These are very easy to

*For notation see pp. 632-633.

work out, as with little practice the square root can be mentally extracted to two places. The area of each flange is half the web area in Division I, is one-fourth the web area in Division II, and is one-sixth the web area in Division III. The columns giving the total sectional area are given merely for the sake of comparison. Note the comparison at the bottom of the table, showing the relative weights of girders according to the three different specifications.

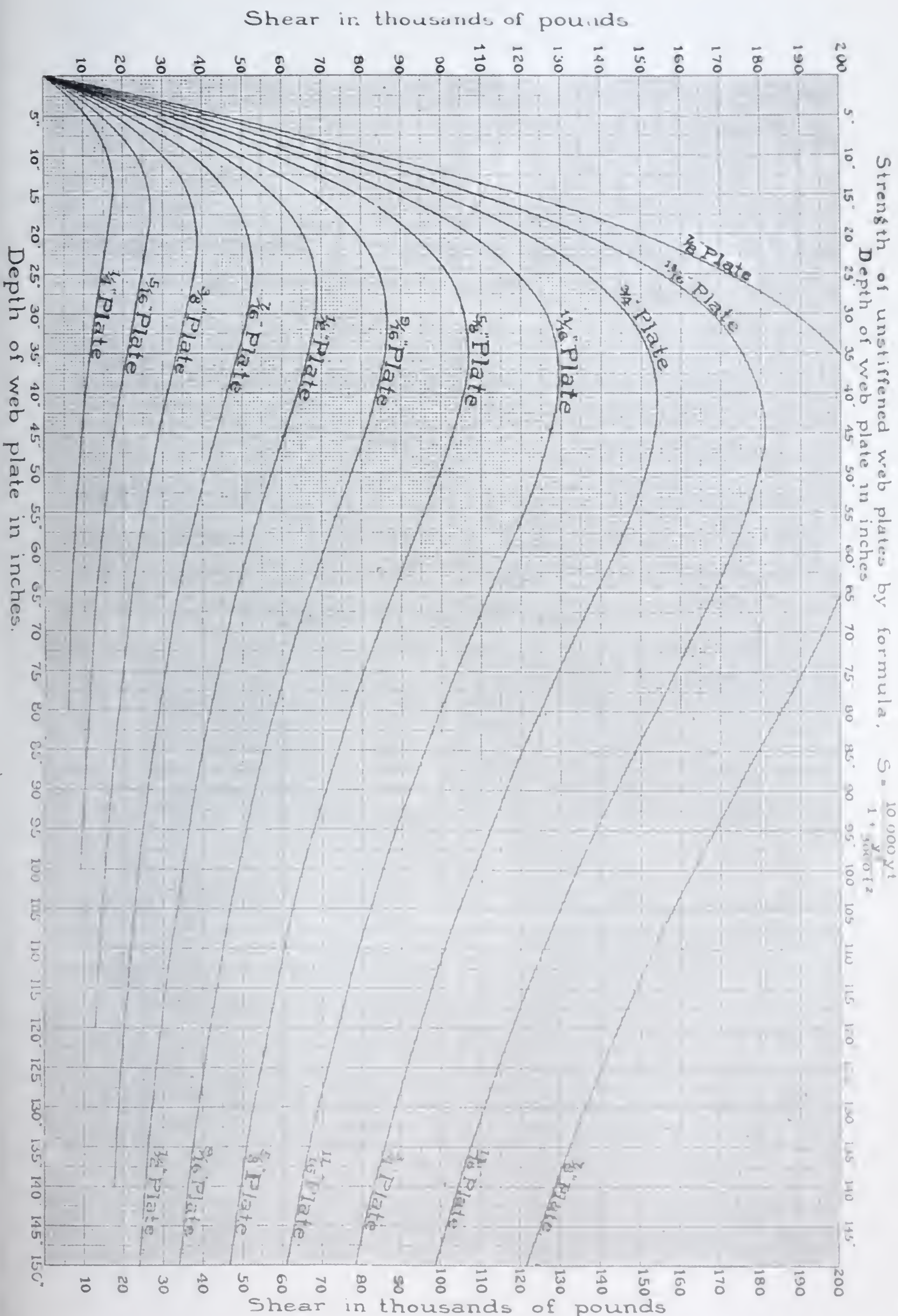
When it is desired to do without stiffeners, the thickness of the web may be determined by any specified formula, and revised if the assumed thickness is not sufficient to take the shear. If the old specification is used—that the allowed shear per square inch on unstiffened webs shall not exceed $\dfrac{10\,000}{1+\dfrac{y^2}{3000t^2}}$—then the thickness of the web can be determined from Chart 1. A little experience enables one instantaneously to pick the minimum thickness with such accuracy that it is not necessary to refigure it after the depth has been determined. It is very often more economical to make the web thick enough to take the shear without stiffeners than to use stiffeners.

TABLE II

ALLOWANCE BEING MADE FOR STIFFENERS AND FILLERS

Class	Division	y	a	a_m	A	A_a	A_x
1	I	$\sqrt{\frac{2M}{fnt}}$	$\frac{ynt}{2}$		$2ynt$		$yt(n+1)$
1	II	$\sqrt{\frac{2M}{ft(n-\frac{1}{4})}}$	$\frac{yt(n-\frac{1}{2})}{2}$		$2yt(n-\frac{1}{4})$		$yt(n+\frac{1}{2})$
1	III	$\sqrt{\frac{2M}{ft(n-\frac{1}{3})}}$	$\frac{yt(n-\frac{2}{3})}{2}$		$2yt(n-\frac{1}{3})$		$yt(n+\frac{1}{3})$
2	I	$\sqrt{\frac{13M}{8fnt}}$		$\frac{8ynt}{13}$		$2ynt$	$yt(n+1)$
2	II	$\sqrt{\frac{13M}{8ft(n-\frac{13}{64})}}$		$yt(\frac{8n}{13}-\frac{1}{4})$		$yt(2n-\frac{13}{32})$	$yt(n+\frac{19}{32})$
2	III	$\sqrt{\frac{13M}{8ft(n-\frac{13}{48})}}$		$yt(\frac{8n}{13}-\frac{1}{3})$		$yt(2n-\frac{13}{24})$	$yt(n+\frac{11}{24})$
3	I	$\sqrt{\frac{211M}{128fnt}}$		$\frac{128ynt}{211}$		$2ynt$	$yt(n+1)$
3	II	$\sqrt{\frac{211M}{128ft(n-\frac{211}{1024})}}$		$yt(\frac{128n}{211}-\frac{1}{4})$		$yt(2n-\frac{211}{512})$	$yt(n+\frac{301}{512})$
3	III	$\sqrt{\frac{211M}{128ft(n-\frac{211}{768})}}$		$yt(\frac{128n}{211}-\frac{1}{3})$		$yt(2n-\frac{211}{384})$	$yt(n+\frac{115}{384})$
4	I	$\sqrt{\frac{7M}{4fnt}}$		$\frac{4ynt}{7}$		$2ynt$	$yt(n+1)$
4	II	$\sqrt{\frac{7M}{4ft(n-\frac{7}{32})}}$		$yt(\frac{4n}{7}-\frac{1}{4})$		$yt(2n-\frac{7}{16})$	$yt(n+\frac{9}{16})$
4	III	$\sqrt{\frac{7M}{4ft(n-\frac{7}{24})}}$		$yt(\frac{4n}{7}-\frac{1}{3})$		$yt(2n-\frac{7}{12})$	$yt(n+\frac{5}{12})$

CHART 1

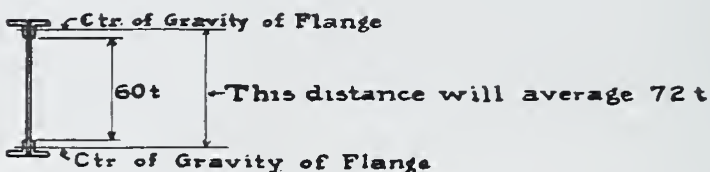

Shear in thousands of pounds
Strength of unstiffened web plates by formula,
Depth of web plate in inches
Depth of web plate in inches.
Shear in thousands of pounds

Passing on to Table II, we have the formulas in which allowance is made for stiffeners and fillers. As before, the forms in Class 1 are the ones which are in most common use. These, while more complicated than the forms in Table I, are very easy of application and, after a little practice, the results are obtained in about the same time as by the ordinary methods.

Another specification for unstiffened webs is that the clear distance between flanges shall not exceed 60 times the thickness of the web. This will result in a girder in which the distance between the centers of gravity of the flanges will average about 72 times the thickness of the web. It is on this basis that Table III is worked out. As this seldom results in an economical

TABLE III

FOR UNSTIFFENED WEB PLATE UNDER SPECIFICATION THAT CLEAR DEPTH BETWEEN FLANGES SHALL NOT EXCEED 60 TIMES THE THICKNESS OF WEB PLATE



Class	Division	y	a	a_m	A	A_a
1	I	$\sqrt[3]{\frac{72M}{f}}$	$\frac{y^2}{72}$		$\frac{y^2}{24}$	
1	II	$\sqrt[3]{\frac{96M}{f}}$	$\frac{5y^2}{576}$		$\frac{y^2}{32}$	
1	III	$\sqrt[3]{\frac{108M}{f}}$	$\frac{y^2}{144}$		$\frac{y^2}{36}$	
2	I	$\sqrt[3]{\frac{117M}{2f}}$		$\frac{2y^2}{117}$		$\frac{y^2}{24}$
2	II	$\sqrt[3]{\frac{1248M}{17f}}$		$\frac{89y^2}{7488}$		$\frac{17y^2}{512}$
2	III	$\sqrt[3]{\frac{2808M}{35f}}$		$\frac{19y^2}{1872}$		$\frac{35y^2}{1152}$
3	I	$\sqrt[3]{\frac{1899M}{32f}}$		$\frac{32y^2}{1899}$		$\frac{y^2}{24}$
3	II	$\sqrt[3]{\frac{60768M}{813f}}$		$\frac{1415y^2}{121536}$		$\frac{271y^2}{8192}$
3	III	$\sqrt[3]{\frac{45576M}{557f}}$		$\frac{301y^2}{15192}$		$\frac{143y^2}{3072}$
4	I	$\sqrt[3]{\frac{63M}{f}}$		$\frac{y^2}{63}$		$\frac{y^2}{24}$
4	II	$\sqrt[3]{\frac{2016M}{25f}}$		$\frac{43y^2}{4032}$		$\frac{43y^2}{2304}$
4	III	$\sqrt[3]{\frac{1512M}{17f}}$		$\frac{y^2}{112}$		$\frac{17y^2}{576}$

$t = \frac{y}{72}$ in all cases

girder, this table is not of very much practical use, but it is presented in order to make the discussion complete.

Another specification for highway bridges which is sometimes used is that the shear per square inch on the web shall not exceed $12\,500 - \dfrac{90y}{t}$. Table IV gives formulas for this speci-

TABLE IV

KETCHUM'S SPECIFICATIONS FOR HIGHWAY BRIDGES CONTAIN THE FOLLOWING RULE FOR WEB PLATES WITHOUT STIFFENERS: SHEAR PER SQUARE INCH ON THE WEB NOT TO EXCEED $12500 - \dfrac{90y}{t}$. THE FOLLOWING TABLE IS BASED ON THIS RULE

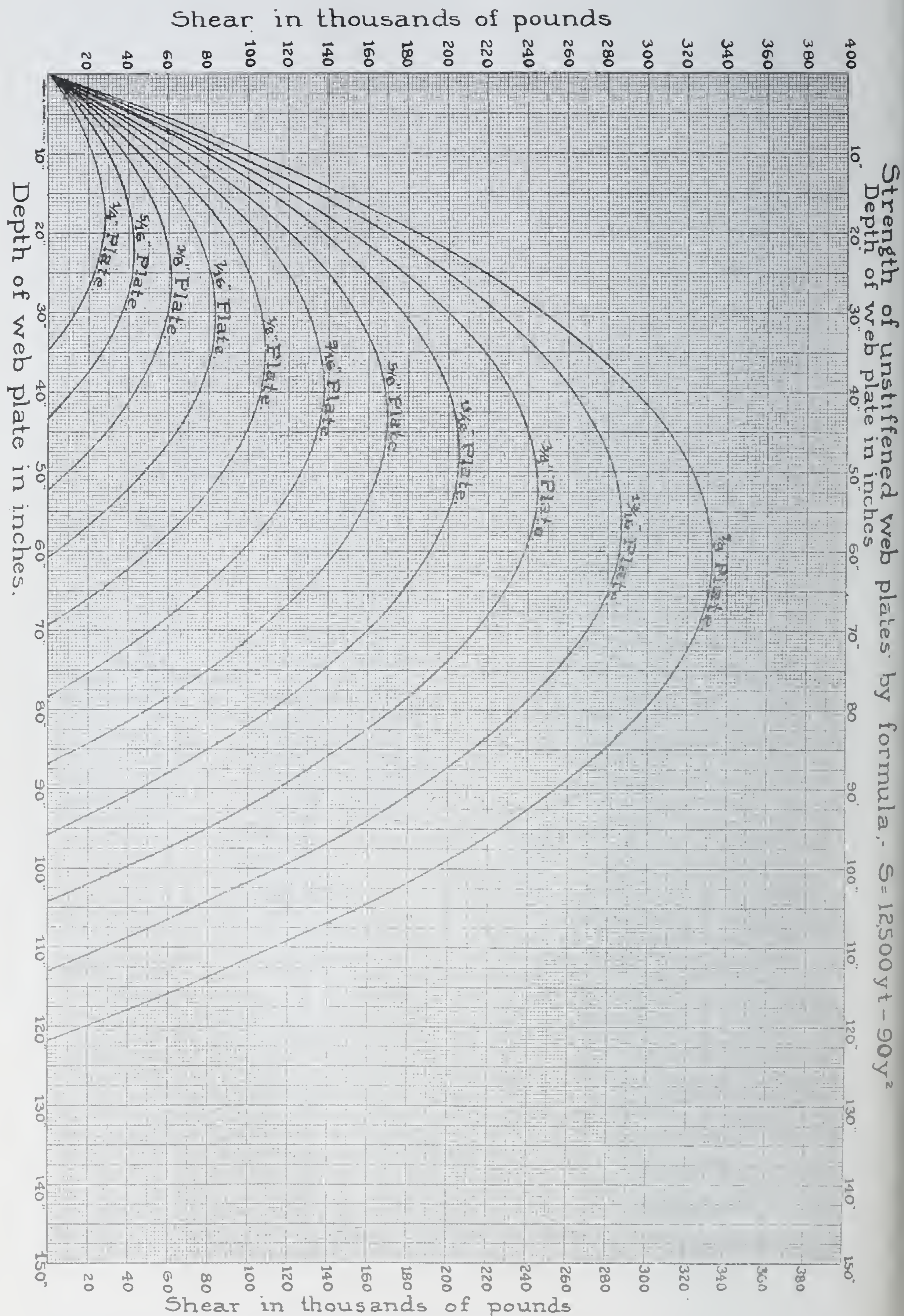
Class	Division	y	a	a_m	A	A_a
1	I	$\sqrt[3]{\frac{1250\,M}{9\,f}}$	$\frac{9y^2}{1250}$		$\frac{S}{12500}+\frac{27y^2}{1250}$	
1	II	$\sqrt[3]{\frac{5000\,M}{27\,f}}$	$\frac{9y^2}{2000}-\frac{S}{100000}$		$\frac{3S}{50000}+\frac{81y^2}{5000}$	
1	III	$\sqrt[3]{\frac{625\,M}{3f}}$	$\frac{9y^2}{2500}-\frac{S}{75000}$		$\frac{S}{18750}+\frac{9y^2}{1250}$	
2	I	$\sqrt[3]{\frac{8125\,M}{72\,f}}$		$\frac{72\,y^2}{8125}$		$\frac{S}{12500}+\frac{27y^2}{1250}$
2	II	$\sqrt[3]{\frac{65000\,M}{459\,f}}$		$\frac{801\,y^2}{130000}-\frac{S}{100000}$		$\frac{51S}{800000}+\frac{1377y^2}{80000}$
2	III	$\sqrt[3]{\frac{3250\,M}{21\,f}}$		$\frac{171y^2}{32500}-\frac{S}{75000}$		$\frac{7S}{120000}+\frac{63y^2}{4000}$
3	I	$\sqrt[3]{\frac{131875\,M}{1152\,f}}$		$\frac{1152\,y^2}{131875}$		$\frac{S}{12500}+\frac{27y^2}{1250}$
3	II	$\sqrt[3]{\frac{1055000\,M}{7317\,f}}$		$\frac{2547y^2}{422000}-\frac{S}{100000}$		$\frac{813S}{12800000}+\frac{21951y^2}{1280000}$
3	III	$\sqrt[3]{\frac{263750\,M}{1671\,f}}$		$\frac{2709\,y^2}{527500}-\frac{S}{75000}$		$\frac{557S}{9600000}+\frac{5013y^2}{320000}$
4	I	$\sqrt[3]{\frac{4375\,M}{36\,f}}$		$\frac{36y^2}{4375}$		$\frac{S}{12500}+\frac{27y^2}{1250}$
4	II	$\sqrt[3]{\frac{1400\,M}{9\,f}}$		$\frac{387y^2}{70000}-\frac{S}{100000}$		$\frac{S}{16000}+\frac{27y^2}{1600}$
4	III	$\sqrt[3]{\frac{8750\,M}{51\,f}}$		$\frac{81y^2}{17500}-\frac{S}{75000}$		$\frac{17S}{300000}+\frac{153y^2}{10000}$

$t = \frac{S}{17500} + \frac{9y}{1250}$ in all cases

fication. This table is a little harder to handle than the first two, but with the help of Chart 2 the writer has found it very convenient. Since this last table requires more adjusting to suit conditions than any of the others, the writer has not been able to get the same results as with the first two as far as speed is concerned.

As has been said, all formulas must be applied with judgment. For instance, if the depth of the girder as determined by the formula is 40 inches, and the required net flange area is 7.5 square inches and, if it is found that the nearest convenient section of flange contains 8 square inches, the depth could be diminished to a depth requiring 8 square inches. Also the depth of

CHART 2


Shear in thousands of pounds
Strength of unstiffened web plates by formula.- S=12,500yt-90y²
Depth of web plate in inches
¼" Plate
5/16" Plate
⅜" Plate
7/16" Plate
½" Plate
9/16" Plate
⅝" Plate
11/16" Plate
¾" Plate
13/16" Plate
⅞" Plate
Depth of web plate in inches.
Shear in thousands of pounds

the web may be juggled to obtain a depth which can easily be obtained from the mills. As will be found if a curve is plotted, using the depth of the girder as the abcissa and the required metal at that depth for the bending moment as the ordinate, this curve will be tangent to a horizontal line at the economic depth. As the curve departs from the tangent slowly for a little distance on either side of the point of tangency, we can see that a few inches difference in the depth near the economic depth will not make any appreciable difference in the weight of the girder.

Waddell, in his "De Pontibus," objects to the use of formulas similar to these on the ground that in long girders the depth becomes too great for convenient handling. With very few exceptions the writer has not found this to be the case in his experience. For girders sustaining uniform loads of a specified amount per foot, the depth obtained by these formulas will increase in proportion to the length, in case we are figuring by the first two tables. For a concentrated load of a fixed amount, the depth will increase as the square root of the length.

Much more could be said on this part of the subject, and many interesting relationships pointed out, but we can not spend more time on this part of the subject and must pass on to the next division.

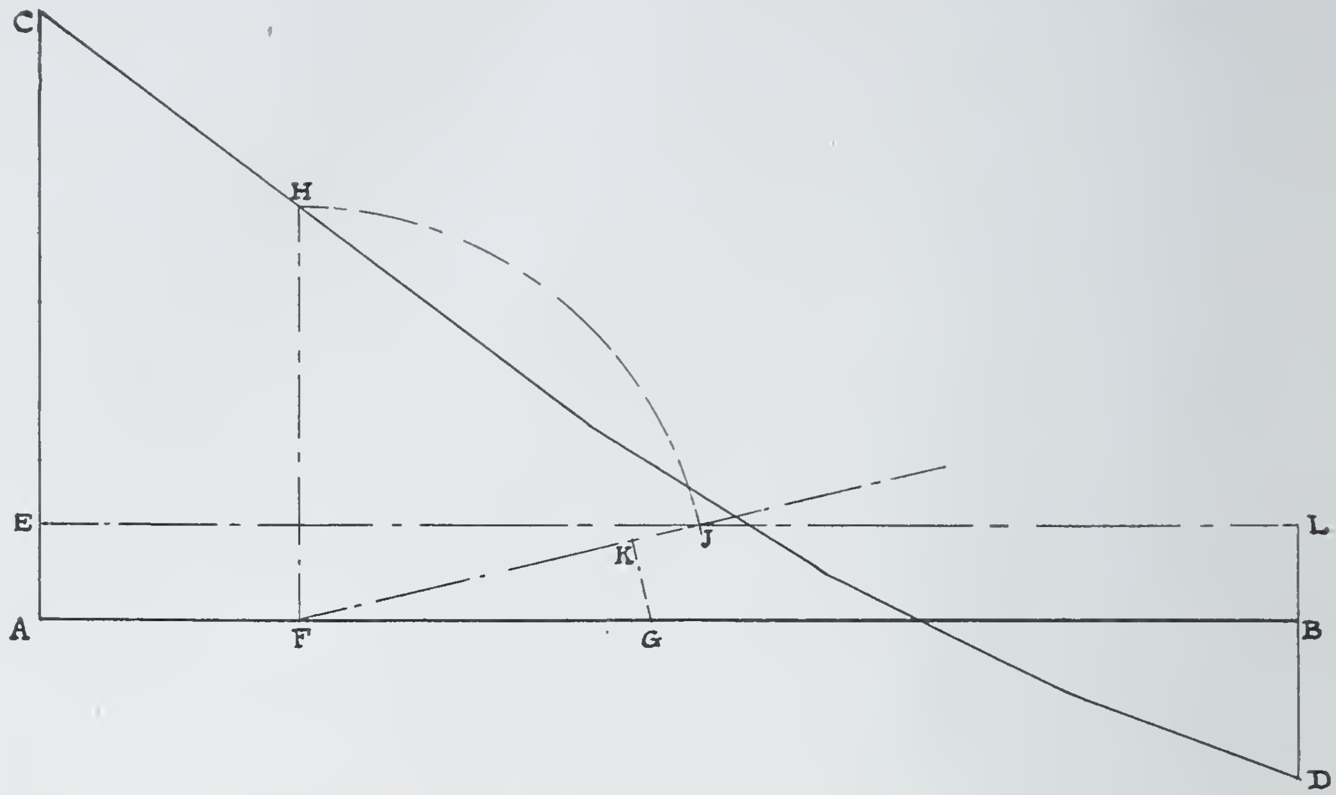
GRAPHIC DETERMINATION OF RIVET PITCH IN FLANGES OF RIVETED GIRDERS

The Common rule for the determination of rivet pitch in the flanges of riveted girders, is that the number of rivets in a space equal to the depth of the girders shall be sufficient to take the shear at the point of the girder in question. There are many ways of stating this, but the above is the substance of all of them.

DIAGRAM 1

GRAPHIC DETERMINATION OF RIVET PITCH IN FLANGES OF PLATE GIRDERS

RIVETS SUBJECTED TO HORIZONTAL THRUST ONLY

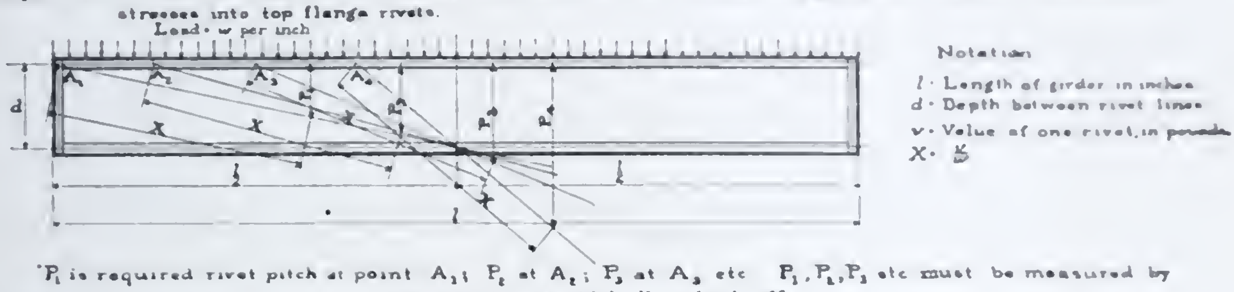


In Diagram 1 is shown a method which is so simple and probably so well known that at first it was not intended to present it, but it is given in order that the discussion may include all cases.

In this diagram let AB represent the length of the girder. Let CD represent the diagram of maximum shear. Lay off AE equal to the value of one rivet on the same scale to which the shear diagram is laid out. Draw EL parallel to AB. At any point, F, where it is desired to determine the required pitch, erect the perpendicular FH. With F as a center and FH as a radius, draw the arc HJ. Draw the line FJ, extending it, if need be, beyond J. Lay off FG equal to the depth of the girder between rivet lines, using any convenient scale, not necessarily the same as that to which the line AB is laid out. Draw GK at right angles to FJ. GK measured on the same scale to which FG was laid off is the required pitch at the point F. This method takes care of the shear only, and does not take account of any vertical load coming on the rivets, as when the top flange is loaded.

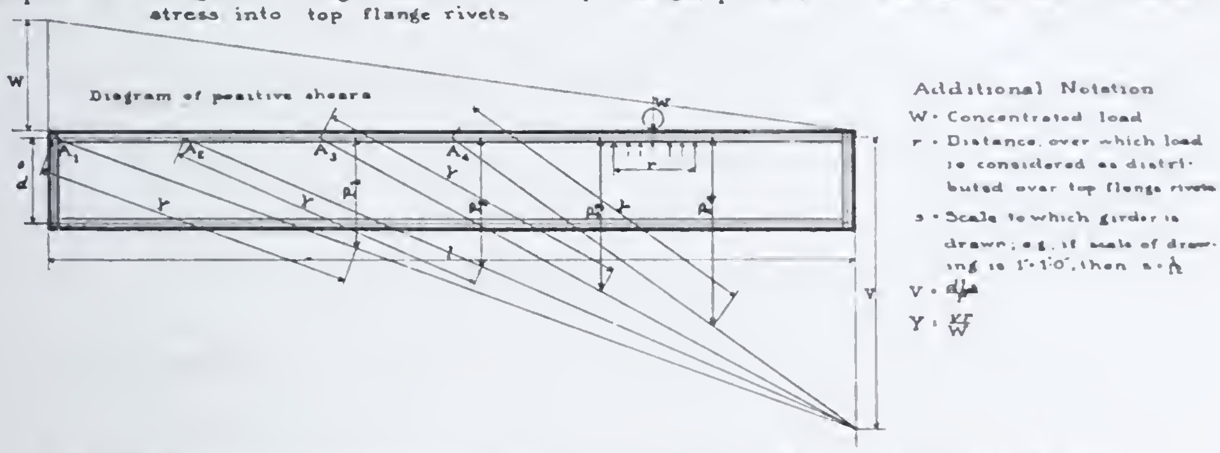
DIAGRAM 2

Example 1 Uniform load of w pounds per inch on top flange, bringing vertical and horizontal stresses into top flange rivets.



P_1 is required rivet pitch at point A_1; P_2 at A_2; P_3 at A_3 etc. P_1, P_2, P_3 etc must be measured by same scale to which X is laid off.

Example 2 Single moving load W on top flange, putting vertical as well as horizontal stress into top flange rivets.



P_1 is required rivet pitch at point A_1; P_2 at A_2; P_3 at A_3 etc. V is to be laid off at full size. P_1. P_2 etc. must be measured by same scale as used in laying off Y.

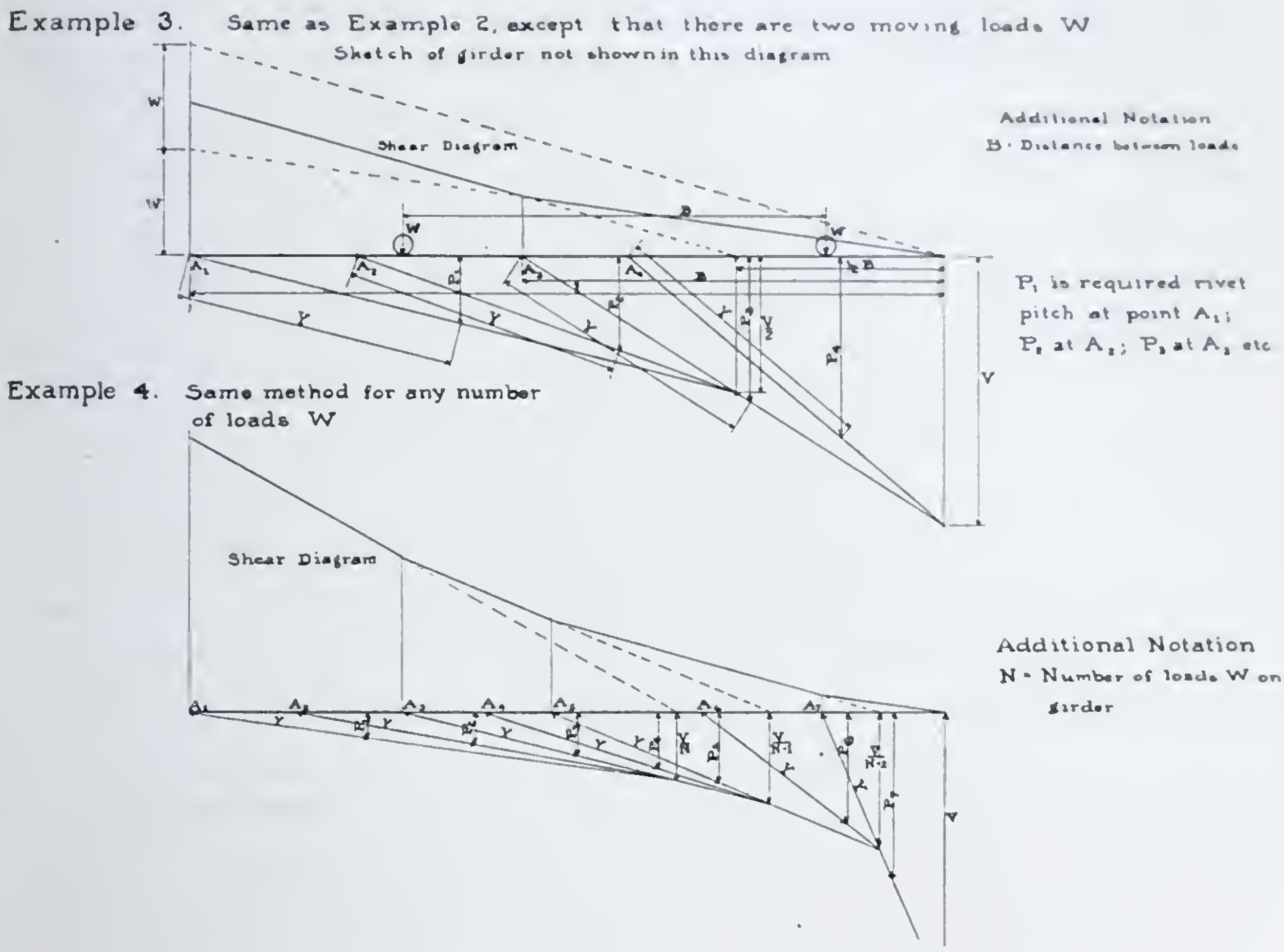
Example 1 (Diagram 2) shows the method of determining the required pitch in a girder under a uniform load of w pounds per inch on the top flange, thus bringing the flange rivets into vertical as well as horizontal shear. In this case, lines are drawn from the points where the rivet pitches are to be determined through the middle point of the opposite rivet line (as shown in the diagram, where the required pitches at the points A_1, A_2, A_3, and A_4 are to be determined, and the lines are drawn through these points and through the middle point of the opposite rivet line) On these lines we lay off the distance denoted by X in the diagram, the value of which is equal to the strength of one rivet divided by the load per inch. For instance, if the value of one rivet is 6000 pounds and the load per inch is 600 pounds, then X would equal 10 inches. This can be laid off on any scale; full size if possible. The vertical distances marked P_1, P_2, P_3, and P_4 are the pitches of rivets required at the points A_1, A_2, A_3, and A_4. These distances are to be measured on the same scale as was used in laying off X.

When we come to concentrated moving loads resting on the top flange, we apply the same principle. This principle, briefly stated, is that if the reciprocals of the resultants at different points of a constant vertical force and a horizontal force varying uniformly, are plotted as ordinates, the abcissas being the positions of the points where the value of the horizontal force is taken, then the resulting curve will be such as would have the equation, $y = \dfrac{AB}{\sqrt{B^2 + x^2}}$

In Example 2 (Diagram 2) we have the diagram of maximum shears representing a uniformly varying force, and the weight as a constant vertical force. At the end of the girder we erect a perpendicular as shown by the line marked V. The value of V is equal to the depth of the girder between rivet lines, multiplied by the length of the girder in inches, multiplied by the scale on which the girder is laid out—that is, the scale is in ratio, not in inches per foot; a scale of one inch = one foot represents a scale of one-twelfth. The value obtained by these multiplications is divided by the distance over which the concentrated load is considered as distributed. The result obtained is the value of V. Through the end of this perpendicular, lines are drawn to the points on the base-line (in this case the rivet line is used as a base-line) where the rivet pitch is to be determined. On these lines the value Y is laid off equal to the value of one rivet multiplied by the distance over which the load is distributed. and divided by the concentrated load. The rivet pitch at the various points is determined in the same way as in Example 1.

In Example 3 (Diagram 3) we have the same thing, except that we are considering two loads instead of one. In this case the perpendicular is erected at the point where the shear line would cross the base-line if it were continued in the same direction in which it is lying at the point where the rivet pitch is to be determined. In the case of two equal loads, this point is one half the wheel-base from the end of the girder. Of course, as soon as one wheel passes off the girder, the diagram becomes the same as explained in Example 2.

DIAGRAM 3



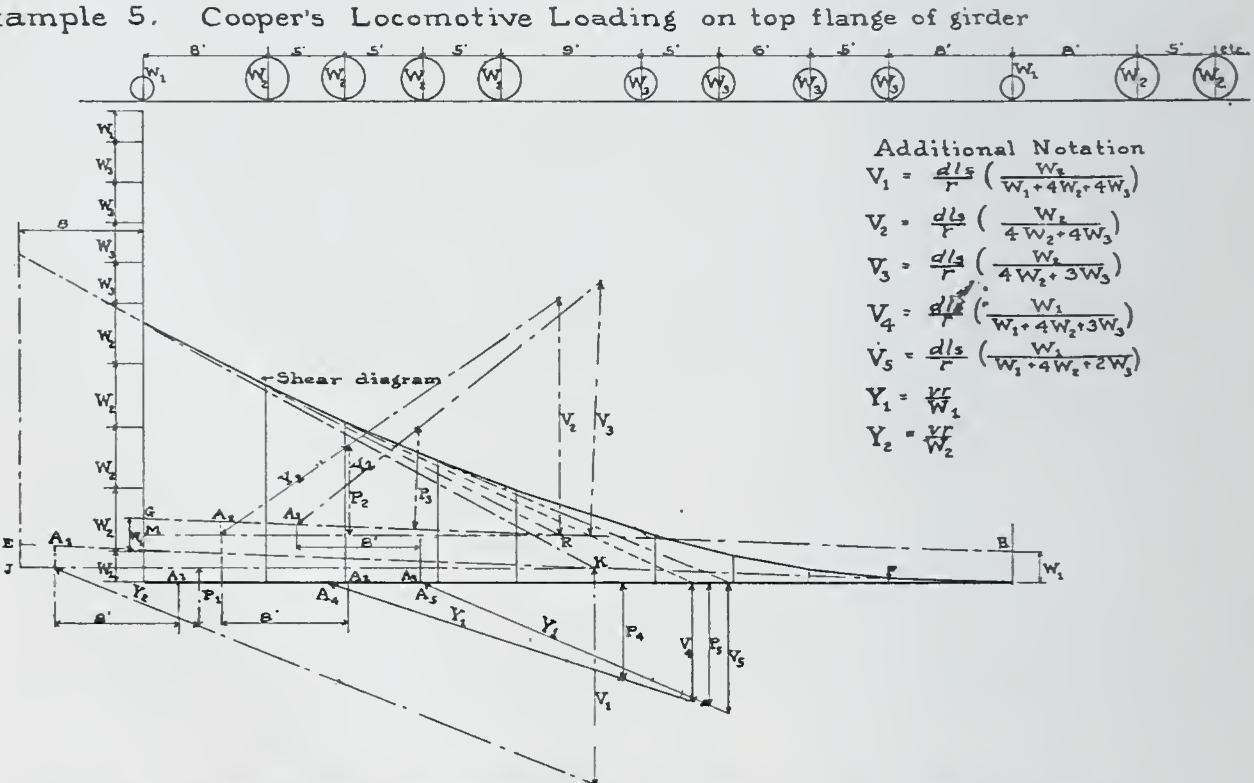
The value of the perpendicular line is one half of V for the space over which two loads are on the girder. When one wheel passes off, the other wheel is a distance equal to the wheel-base from that end. It is obvious then, that a line passing through a point the distance of the wheel-base from the end of the girder and through the perpendicular erected at half the wheel-base distance from the end, intersecting this perpendicular at a point one half V from the base-line will, if continued, intersect a perpendicular drawn from the end of the base-line, at a point a distance of V from the base-line. This principle holds good all through this system—that wherever one load passes off the girder, the diagram automatically changes to the diagram for the loading actually on the girder. Of course, it is not necessary to carry the diagram past the center-line of the girder unless there are peculiar circumstances such as the writer has never encountered in his experience.

Example 4 is the same method applied to any number of equal loads. In this case, as before, the perpendiculars are erected at the points where the shear line would intersect the base-line if continued. The length of these perpendiculars is always V divided by the number of loads on the girder when the point under consideration is under maximum shear. The value Y is affected only by the weight of the concentrated load, the distance over which it is distributed among the rivets, and the strength of a rivet. It is not affected by the number of loads on the girder.

So far we have considered only loads which are uniform. It often happens that the loads vary as in Cooper's locomotive loading. Example 5 (Diagram 4) shows this loading. In this case, the greatest stress on rivets in the eight feet at the end, occurs when the first wheel is off the girder. The rivets at points more than eight feet from the end are subject to two conditions—the first, when the end load is over the point, in which case the horizontal shear is probably greater but the vertical shear less than in the second case, when the first of the heavy loads is over the point. The pitch should be determined for both cases and the

DIAGRAM 4

Example 5. Cooper's Locomotive Loading on top flange of girder



V_1 & V_2 could be drawn at right angles to EF or GH in same manner as V_3, in which case, horizontal lines JK or MR would not be needed; P_1 & P_2 would be drawn at right angles to EF and GH.

Diagram shown in dot and dash lines is shifted 8′ to left from main diagram A_1, A_2 & A_3 represent points under W_2; A_4 & A_5, under W_1. Pitch for any point should be determined for both conditions and smaller pitch used

minimum pitch taken. In the case of the eight feet at the end, we resort to the familiar method of shifting the shear diagram eight feet to the left, as shown by dotted lines in the diagram. The dotted diagram is inclined, so, if the values V_1, V_2, V_3, etc., are drawn vertically, the measurements must be taken to horizontal lines as shown by the lines KJ and RM; or the lines may be drawn at right angles to the inclined base-line of the diagram—the latter, while not absolutely accurate, being generally the preferable method, the error usually being inappreciable. In the diagram, the lines V_1, V_2 are drawn by the first method and V_3 by the second method. For all points farther than eight feet from the end, the weight of the end load must be deducted from the shear shown by the dotted diagram. In the case of the pitch required for the end load we use the methods before described, except that the value of the vertical lines is the value called V in the previous diagrams, multiplied by the load producing the vertical stress in the rivets, divided by the total load on the girder. This principle holds for all conditions of loading—that the value of the vertical lines shall represent the value we have called V, which is the depth of the girder in inches, multiplied by the length in inches, multiplied by the scale of the drawing, divided by the distance over which the vertical stress is considered as distributed; the value, V, obtained in this way to be multiplied by the weight producing the vertical shear in the rivets, divided by the total load on the girder when the point under consideration is under maximum shear.

So far, we have considered only live loading in the case of concentrated loads, but the same principles hold good in the case of combined loads of any description. In the case of Example 6 (Diagram 5) we show a combination of dead and live loading, in which part of the dead load is resting on the top flange—thus helping to produce vertical shear in the rivets—and part is supported otherwise. The shear diagram is drawn in the usual manner for the combined loads. For the weight producing the vertical shear in the rivets we use the concentrated weight plus the portion of the distributed dead load resting on the top flange within the limits of the space over which the concentrated load is considered as distributed over the rivets. This load we desig-

DIAGRAM 5

Example 6. Combination of dead and live loading

Assumed loading: Dead load – w_1 per inch resting on top flange and w_2 per inch supported otherwise.

Live loading – Three concentrated loads W

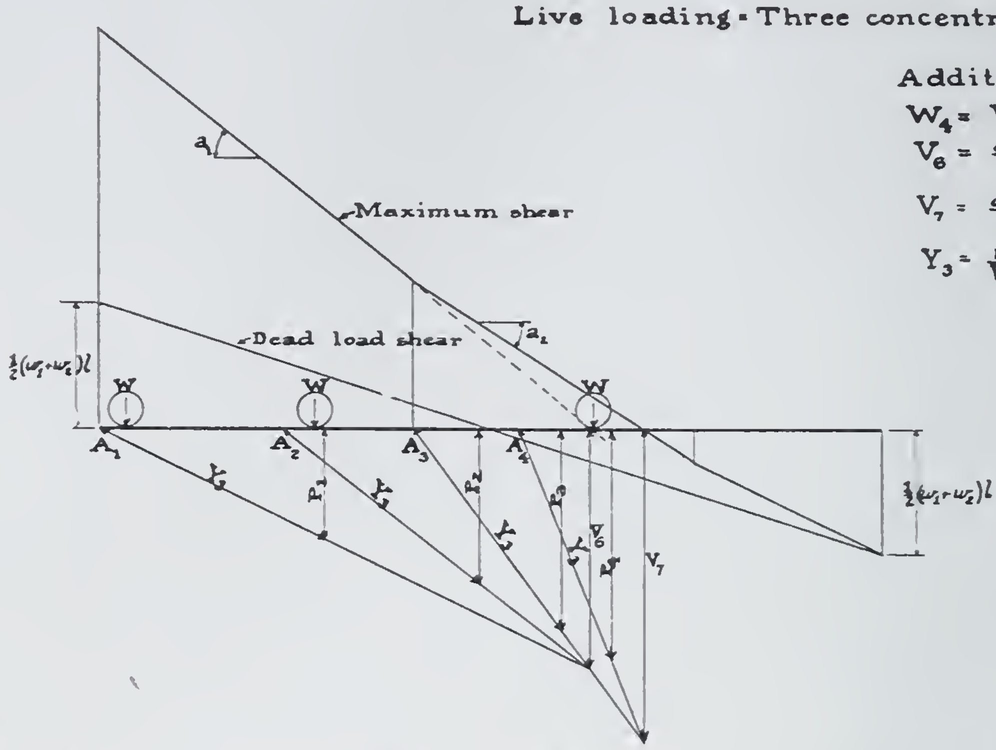
Additional Notation

$W_4 = W + w_1 r$

$V_6 = \frac{dl_s}{r}\left(\frac{W_4}{l \tan a_1}\right)$

$V_7 = \frac{dl_s}{r}\left(\frac{W_4}{l \tan a_2}\right)$

$Y_3 = \frac{vr}{W_4}$



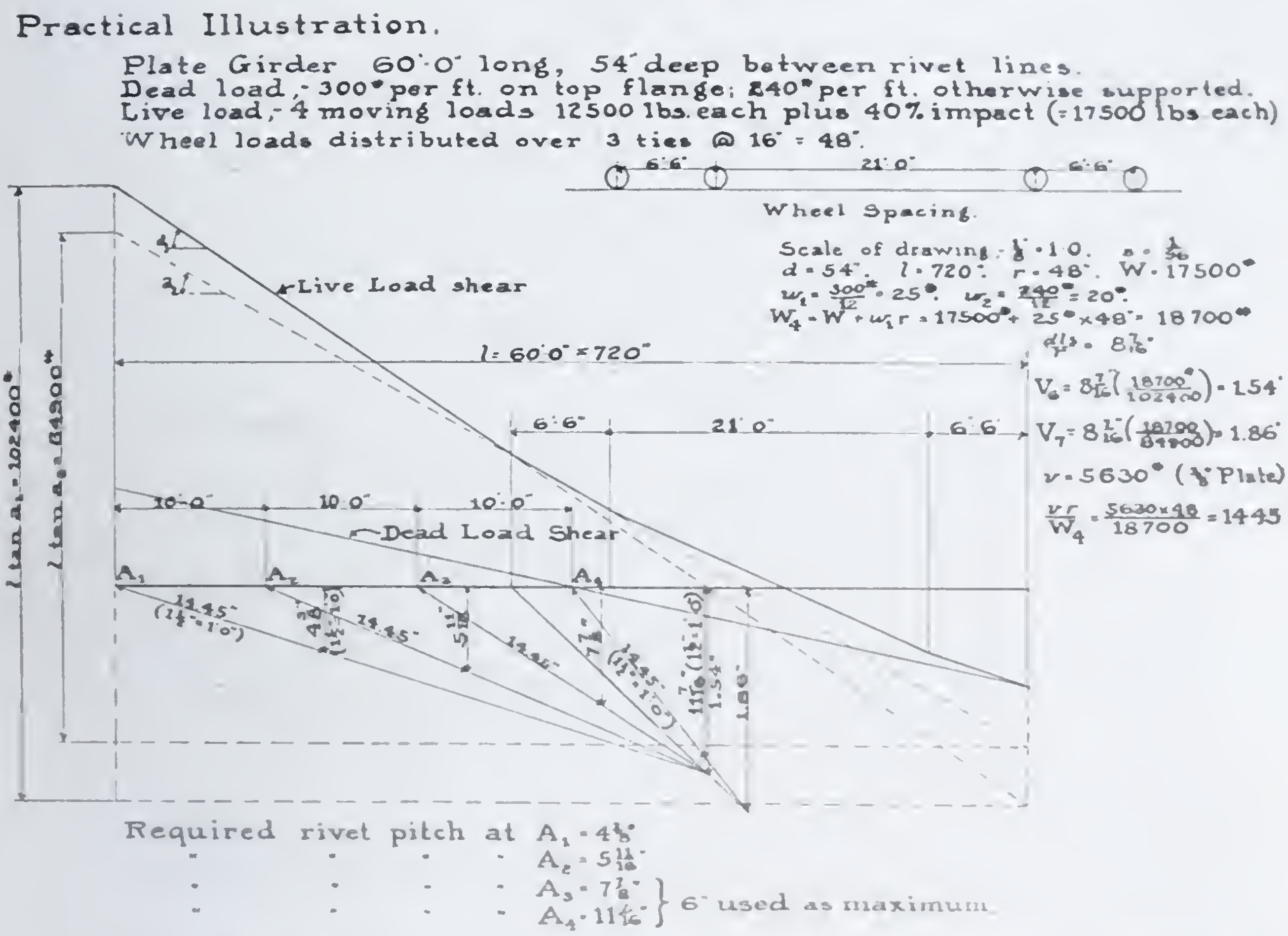
P_1 is required rivet pitch at point A_1, P_2 at A_2 etc. V_6 and V_7 are to be laid off at full size. Measure P_1, P_2 etc. by same scale as used in laying off Y_3.

nate as W_4 in the diagram. The verticals are again drawn from the points where the shear diagram would become zero if continued in the same direction. The values of the verticals are our value previously designated as V, multiplied by the weight, W_4, divided by the total load on the girder at the time when the point we are considering is under the maximum shear. It must be borne in mind that only one of these values of V needs to be figured, as when one load after another passes off the girder the value of the vertical will automatically change to the next required value simply by passing the line from the point where the end load is, when another load is passing off the girder, through the point in the vertical we have been using; producing it till it cuts the vertical line in the next required position, which will give us the next value of V.

The question always arises, in connection with a method of this kind, as to whether it saves any time; and then the next question is as to whether it is so difficult to remember as to limit its usefulness. To the first question we would answer decidedly in the affirmative. Our experience has been such as to lead us to say that results can be obtained by this method in about one-

fourth the time required for any other method known to the writer. As to the second question, we would answer that the formulas are no harder to remember than the common formulas for bending moments in simple beams. The principles for determining the values of V have been stated before, while the values of Y are simply the pitch which would be required if we had only the vertical shear without the horizontal.

DIAGRAM 6

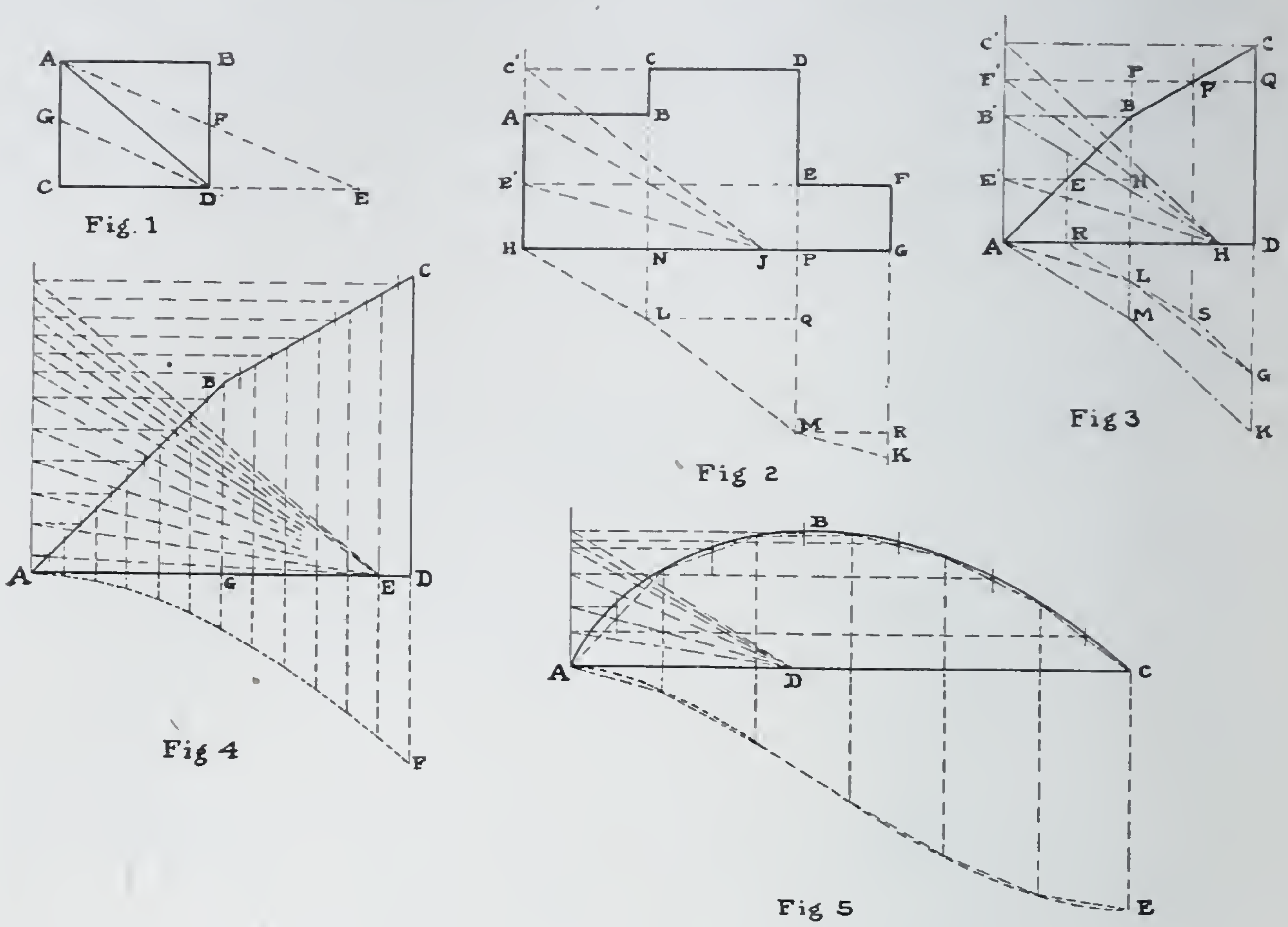


In Diagram 6, the writer presents a practical illustration taken at random from his note book. This girder was for a 60-foot span, supporting a wooden floor and one rail of a track. The live loading is an interurban car such as is used on the Charleroi Division and on the Washington and Canonsburg Division of the Pittsburgh Railways Company, allowing 40 per cent. of live load for impact. The details of the application of the method are worked out on the diagram and do not need any further explanation.

There are many other interesting points which might be brought out with regard to the application of graphics in connection with this subject, but it is impossible to exhaust the subject in the course of one paper, so we must pass on to our next division.

DIAGRAM 7

PRINCIPLES OF GRAPHIC INTEGRATION OF AREAS



PRINCIPLES OF GRAPHIC INTEGRATION APPLIED TO BEAMS AND ARCHES

In Diagram 7 we give the basic principles of graphic integration. These are well known, but we are reviewing them here in order to refresh your memories if, as is probably the case with several of those present, your line of work has not been such as to keep you in touch with all the mathematical aspects of this subject.

In Fig. 1 (Diagram 7) we have the rectangle ABDC. The area is of course AB times AC. Now if we select some other point on the line CD produced, as at E, and draw the line EA cutting BD at F; then CE : AC :: AB : BF, hence CE $\times$ BF $=$ AB $\times$ AC. If from D we draw a line parallel to AE, cutting AC at G; then GG would be the same as BF and CE $\times$ CG $=$ AB $\times$ AC. As each ordinate from the line AB to the line AF is proportional to the distance of the ordinate from A, it is obvious that the ordinate at any point multi-

plied by CE is equal to the area of the part of the rectangle to the left of the point where the ordinate is taken. In Fig. 2 (Diagram 7) we have a figure, ABCDEFGH, in which all the angles are right angles. If we choose any point, J, on the line GH and draw rays from that point to points on the line AH or AH produced, these points being projections from the different corners of the figure, then by the same reasoning, if we draw the line HK, the different parts of which are parallel to the corresponding rays—since JH $\times$ NL is the area of HABN; QM $\times$ JH is the area of NCDP; and RK $\times$ JH is the area of PEFG—the ordinate at any point from this line to the line HG multiplied by JH will be equal to the area of that part of the figure to the left of the point chosen, and GK $\times$ JH will be the area of the whole figure. But if the figure is not rectangular but is bounded by straight lines—as ABCD, Fig. 3 (Diagram 7)—then, by projecting upon the vertical line through A the mid-points (E and F) of the sides AB at BC, respectively; drawing rays to these projected points from an assumed pole, H; and constructing parallels to these rays, in the manner previously indicated, we get the broken line ALG. This line will be a series of chords to the true integral line, which will be a curve—since DG $\times$ AH is the area AE'NPQD which is equal to ABCD. The ordinate, projected below the point B from the line AD to AG, multiplied by AH will be the area of that part of the figure to the left of the point B. The ordinate DG multiplied by AH will be the area of the whole figure. At no other point, however, will this be the case. In this figure we have shown an incorrect method given by at least one author. If the points B and C are projected on the vertical line through A; the rays shown by the dot-and-dash lines are drawn; and from them the line AK is developed—then, this author claims, we will have a series of tangents to the curve of integration. Granting, for the moment, his contention we would still call attention to the fact that a series of chords is a more accurate basis on which to plot a curve than a series of tangents. This fact hardly needs a demonstration. But, besides this, the method does not produce a series of tangents. The fallacy of this statement is obvious from this figure. The line AG must be a series of chords as it is based on the average height of the dif-

ferent parts of the figure. Now, if you will look at the drawing you will easily see that the line AK cannot be a series of tangents to a curve of which the line AG is a series of chords. If the author in question had used the mid-points R and S as points of change in his line, then he would have had a series of tangents.

If it is desired to plot the true curve, this may be done by dividing the figure into a number of segments as is done in Fig. 4 (Diagram 7) and projecting the mid-point of each segment to the vertical line through A and proceeding as before. By the use of a "curly triangle"—as a Frenchman of the writer's acquaintance called it—the true curve can be very accurately drawn.

In case the figure is not bounded by straight lines, as in Fig. 5 (Diagram 7), we may divide it into segments as before, then draw a series of chords to the curve, as shown, and at the mid-point of these segments draw short vertical lines between the chords and the curve. At points two-thirds of the distance from the chord to the curve, project lines over to the vertical line through A and proceed as before to draw the line of integration. The results by this method will correspond to those obtained by the use of Simpson's rule and the writer has found them very accurate..

At the time that the writer worked out the application of this principle to determining the deflection of beams, he did not know that others had preceded him in this investigation. Among those might be mentioned Professor O. Mohr, Professor H. K. Thayer of our own Society, Professor M. S. Ketchum, and Professor W. K. Hatt. However, as our investigation has been along a little different line from the line followed by these other gentlemen, we present it here, not wishing in any way to detract from the credit due to them or to others whose names may not be known to the writer.

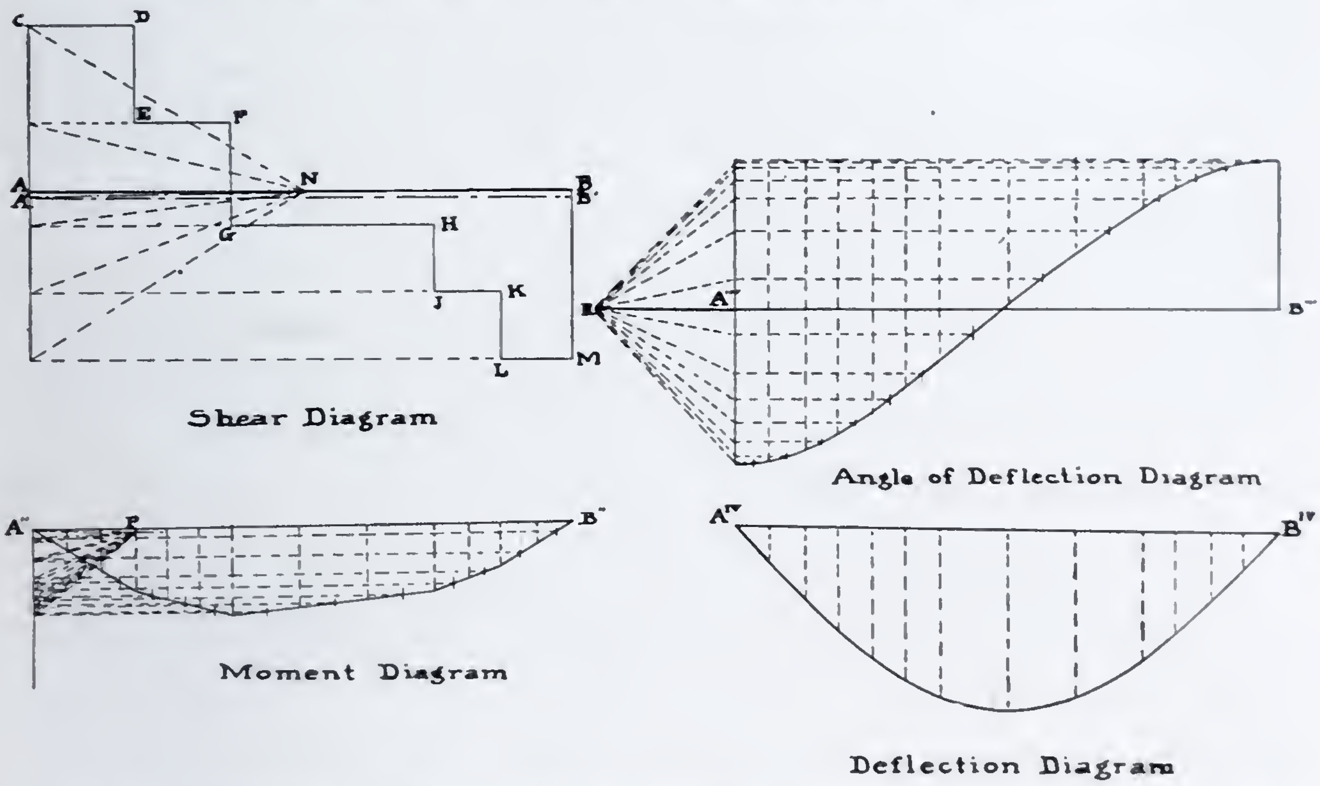
The basic principle of this method is the fact that the shear diagram represents the integral of the load diagram: the moment diagram the integral of the shear diagram; the angle-of-deflection diagram the integral of the moment diagram; and the deflection diagram the integral of the angle-of-deflection diagram. In other words, if we let $L =$ the load; $S =$ the shear; $M =$ the moment; $A =$ the angle of deflection; and $D =$ the deflec-

tion, we will have the following equations: $S = \int Ldx + C$; $M = \int Sdx + C$; $A = \int \frac{Mdx + C}{EI}$; and $D = \int Adx + C$.

With the first of these equations we will not spend time, as the shear diagram is more easily developed without reference to it.

DIAGRAM 8

GRAPHIC INTEGRATION APPLIED TO BEAMS
SIMPLE BEAMS WITH UNIFORM MOMENT OF INERTIA



In Diagram 8, we show the application of this principle to simple beams with a uniform moment of inertia. The line CDEFGHJKLM represents the shear diagram. As has been said, this represents the integral of the load diagram. The constant of integration is not yet known. This is elementary work, but we are going over it to show the relationship to what follows. We select a pole at N and draw a force diagram and then the equilibrium diagram by the usual method. As we know that the constant of integration in this case is zero—since the bending moment at the end is zero—we draw the closing line to the moment diagram—the line A″B″—which in this case is not exactly horizontal. A line drawn from N in the shear diagram parallel to A″B″ in the moment diagram will cut the load line at

A′. The horizontal line A′B′, then, is the true closing line of the shear diagram, and A′C is the constant of integration for the shear diagram.

The moment diagram is divided into segments and the mid-points of those segments projected to the vertical line through A″. The projecting lines are to be parallel to A″B″. A pole, P, is selected, which should be in the same horizontal line with A″, and the force polygon is drawn in the same manner as for the shear diagram. By the usual method we draw the equilibrium polygon as before, giving us the line of the angle-of-deflection diagram. Here, again, we do not know what the constant of integration is. However, we project the mid-points of the segments, into which the diagram is divided, to the vertical line at the left of the diagram and choose a pole, R, as in the case of the other diagrams. Drawing the rays of the force polygon, we develop the deflection diagram as an equilibrium diagram. As we know that the constant of integration in this case is zero, we draw the closing line $A^{iv}B^{iv}$. From R we draw the line RA‴ parallel to the closing line of the deflection diagram. We then draw the horizontal line A‴ B‴, which is the closing line of the angle-of-deflection diagram.

As you all know, to find the moment at any point, we multiply the vertical ordinate of the moment diagram by the value in pounds of the pole NA (or, to be more exact, the horizontal component of the pole). In handling this method, one should use the inch instead of the foot as the standard of measurement, as moments of inertia are nearly always given in inches and the modulus of elasticity is generally given in relationship to the square inch and, unless the operator is very careful, the use of foot-pounds instead of inch-pounds in designating the bending moment is likely to lead to confusion.

The tangent of the angle of deflection at any point is found by multiplying the ordinate of the angle-of-deflection diagram from the curve to the closing line at that point by the pole A″P, multiplying the result by the pole AN and dividing the result by the moment of inertia multiplied by the modulus of elasticity. We will not take time to discuss how the *EI* comes in in this equation. That can be found in any work on structural mechanics.

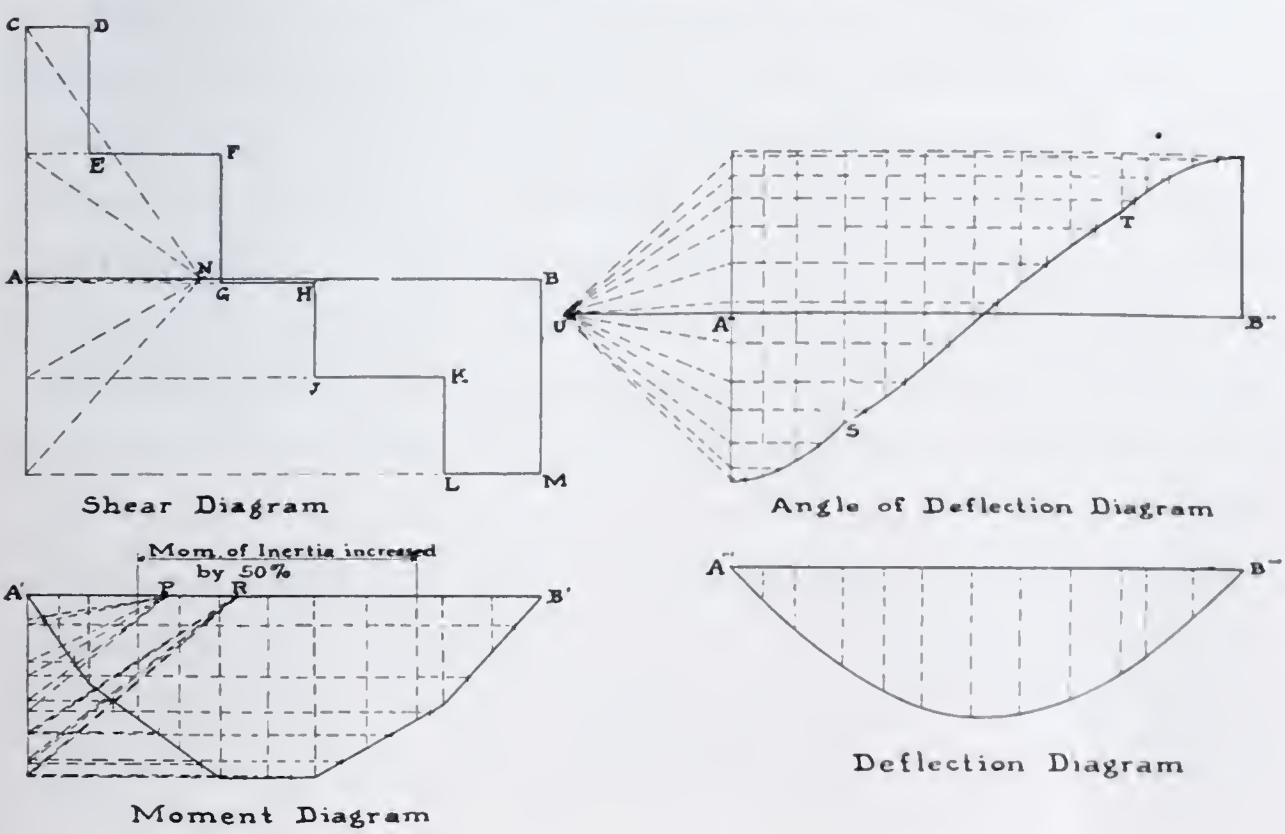
The deflection at any point may be found by multiplying the ordinate of the deflection diagram at that point by the pole RA‴ and the result by the product of the pole A″P and the pole AN and dividing the result by the *EI* of the beam.

Where the loading is simple and it is necessary to know only the maximum deflection, this method is more cumbersome than the method of calculation, and it may be that this is the reason for the neglect of the graphical method. But where the loading is complicated, and where—as in the case of heavy machine supports—it is very often necessary to know the deflection at points other than that where the maximum occurs, the writer has found that the graphical method will produce accurate results in from one-tenth to one-fourth the time required for calculation.

All the published formulas on deflection which the writer has seen, have been based on the assumption that the moment of inertia is constant throughout the length of the beam. It has been pointed out by writers on the subject that the graphical method lends itself very readily to the determination of deflec-

DIAGRAM 9

GRAPHIC INTEGRATION APPLIED TO BEAMS
SIMPLE BEAMS WITH VARYING MOMENTS OF INERTIA



tion of beams where the moment of inertia is not constant. Not all of the writers take the trouble to explain the means of applying the method to these cases, and so rob the method of one of its chief charms—that is, its adaptability to varying circumstances. In Diagram 9, we show a case of the same kind which we have just explained, except that the moment of inertia is varied. Simply for convenience we have assumed that the moment of inertia is increased by 50 per cent. in the central portion as indicated in the diagram. The variation may occur at any point or at any number of points. It may be in any proportion, but we have chosen this ratio for ease in explanation.

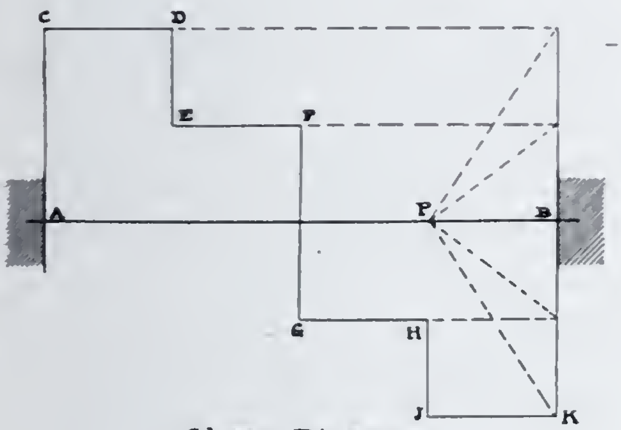
In this case the procedure is the same as explained for the beam with a constant moment of inertia, except when it comes to developing the angle-of-deflection diagram. A pole, A′P, is chosen in the moment diagram, and a pole 50 per cent. greater, A′R, is laid off for use in the portion of the beam having the increased moment of inertia. You will notice that there is a kink in the angle diagram at the points S and T which correspond to the points where the inertia changes. It should be remembered when determining the tangent of the angle of deflection, or the deflection, that, whichever pole is used in the multiplications, the corresponding moment of inertia must be used in the divisions.

But there is another line in which this method is useful—that is, in the analysis of fixed beams. The writer has not found any work in which this matter is discussed and we believe that from now on we are passing over untrodden ground. It might be said, by the way, that this method also lends itself to the analysis of continuous beams, but as we had to draw the line somewhere in our discussion, we decided to omit all reference to continuous beams.

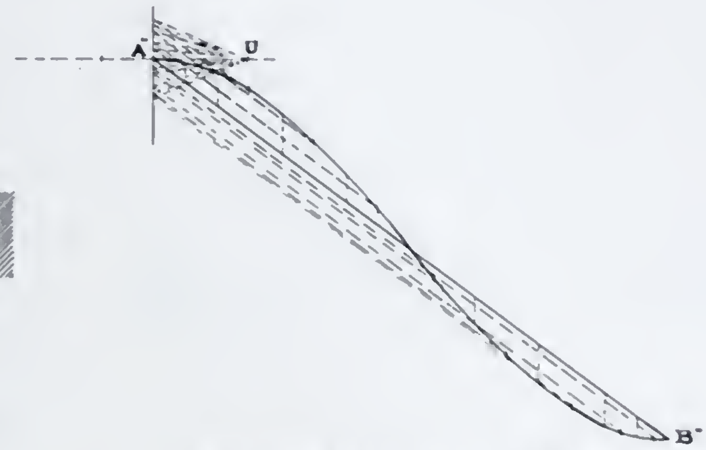
So far as the writer knows, all the published formulas for finding the stresses in fixed or continuous beams are based on the assumption that the moment of inertia of the beam is a constant quantity. Where this is not the case, the formulas are absolutely worthless.

DIAGRAM 10

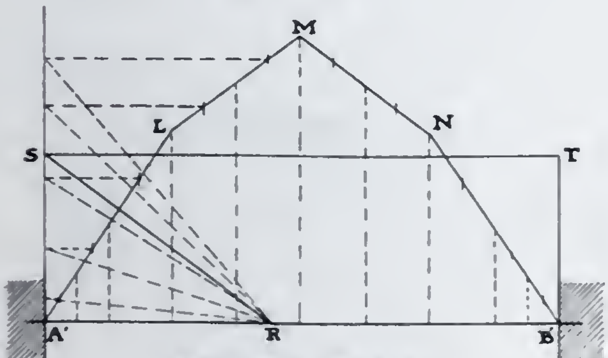
GRAPHIC INTEGRATION APPLIED TO BEAMS
FIXED BEAMS WITH UNIFORM MOMENT OF INERTIA
SYMMETRICAL LOADING



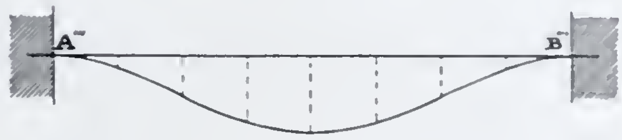
Shear Diagram



Angle of Deflection Diagram



Moment Diagram

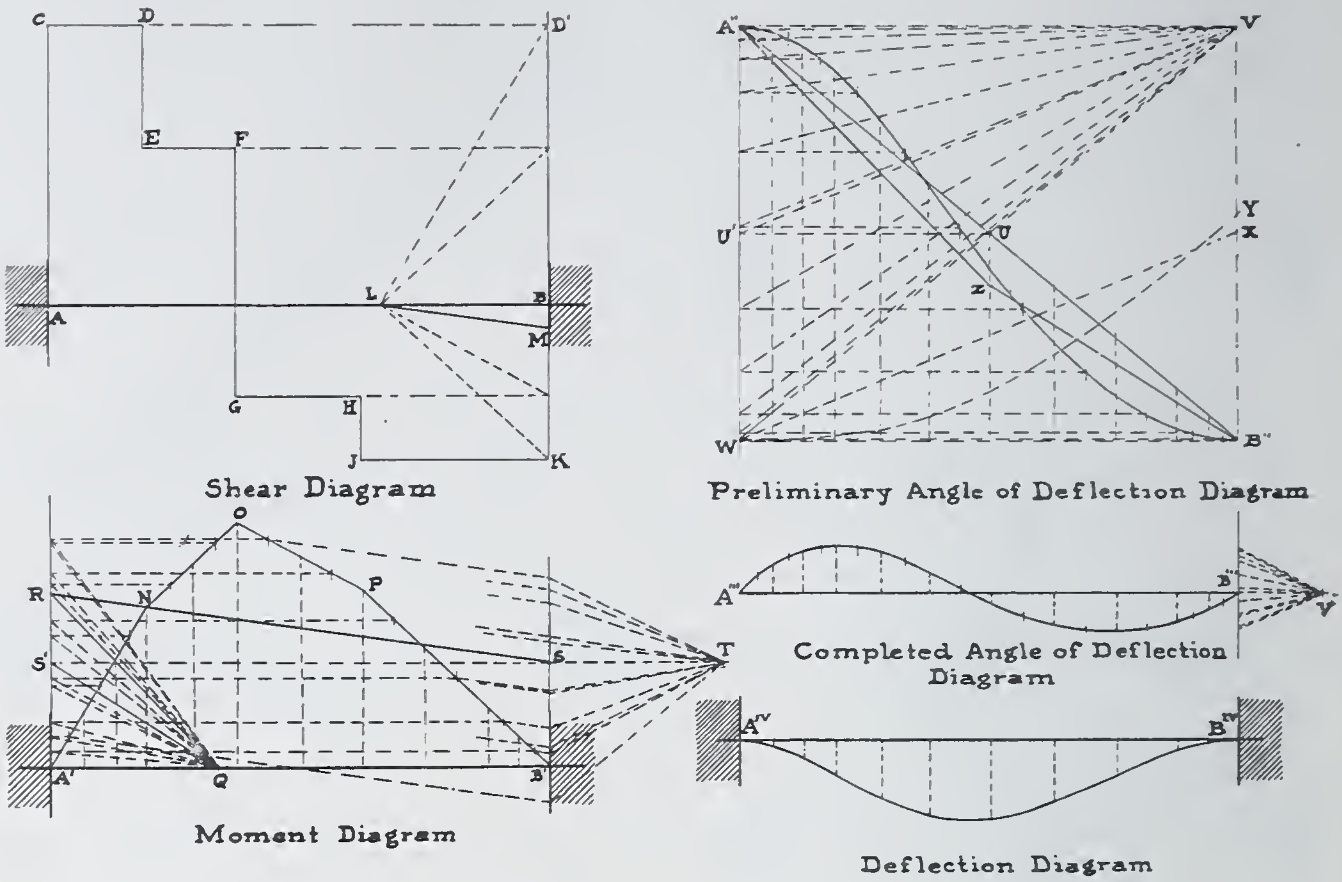


Deflection Diagram

In Diagram 10 we show a fixed beam with a uniform moment of inertia and symmetrical loading. This is the simplest case. In this case we proceed as if we were analyzing a simple beam. The moment diagram is drawn without reference to the fact that it is a fixed beam. The angle-of-deflection diagram is drawn as shown by the curved line A″B″. In this case, however, the constant of integration for the angle diagram is zero, so the closing line is drawn from A″ to B″. From the pole R in the moment diagram we draw the line RS parallel to the closing line of the angle diagram. We then draw the horizontal line ST which completes the moment diagram. Note that in projecting the mid-points of the segments of the angle diagram to the vertical line through A″, they are projected by lines parallel to the line A″B″. The deflection diagram is developed in the usual way.

DIAGRAM 11

GRAPHIC INTEGRATION APPLIED TO BEAMS
FIXED BEAM WITH UNIFORM MOMENT OF INERTIA
UNSYMMETRICAL LOADING



In Diagram 11 we have a fixed beam with a uniform moment of inertia, but unsymmetrical loading. In this case the moment diagram is developed in the usual way without reference to the fact that the beam is fixed. A pole, Q, is chosen and a preliminary angle diagram drawn as shown by A″B″. The closing line in this case will not be a straight line. Two conditions must be met—namely, that the angle of deflection must be zero at both ends, and that the deflection must be zero at both ends; a straight closing line would answer the first but not the second. In order to meet the first condition, the closing line must extend from A″ to B″ and it can be either straight or curved. In order to meet the second condition, the area A″VB″ bounded by the angle-of-deflection curve must be equal to the area A″VB″ bounded by the closing line. In order to determine the form of the closing line we integrate the angle diagram, using the point V as a pole, which gives us the line WY. We then integrate the area A″VB″U, by projecting the mid-point U over to U′, and parallel to VU′ we

draw WX. Now, since B″Y × A″V is the area of the figure A″VB″ bounded by the angle diagram and B″Y × A″V is the area A″VB″U; then XY × A″V, being the difference between the two areas, is the area included between the angle diagram and the line A″UB″. This, then, must be the area between the line A″UB″ and the true closing line, which must be a parabolic curve. The demonstration of this fact is easy when we remember that we are dealing with a triangular area, RS′S, and that the area of a triangle increases as the square of the distance from the apex. Also, note that the shear diagram for a simple beam with a uniform load is a sloping line; and the moment diagram, which is the integration of the shear diagram, is a parabola.

The area of the space between this parabolic curve and the line A″UB″ will be equal to A″V multiplied by two-thirds of the distance from the point U to the point where the parabolic curve would cut the line UZ. This distance from U to the point where the curve would cut the line UZ is, then, 1½ times XY. But, since the tangents to the parabola would cut the line UZ at twice the distance from U that the curve would, we lay off UZ equal to three times XY, and draw the lines A″Z and B″Z which are the tangents to the true closing line.

We draw the line QR in the moment diagram parallel to A″Z, and QS′ parallel to B″Z, and draw the line RS which is the closing line of the moment diagram. The line LM, parallel to RS, divides the load line into D′M—the vertical reaction at the left support—and MK— the vertical reaction at the right support. The mid-points of the segments of the moment diagram are then projected to the line B′S by lines parallel to RS; and a pole, T, in a horizontal line through S is chosen, the force polygon is drawn, and from that the completed angle-of-deflection diagram is drawn. Then, by projecting the mid-points of the segments of that diagram to the vertical line at the left end of the diagram and choosing a pole, V′, in the horizontal line through B‴, we draw the deflection diagram.

These last two diagrams will check the accuracy of our work very closely, as a very slight error in the position of the closing line of the moment diagram will show up very decidedly in one

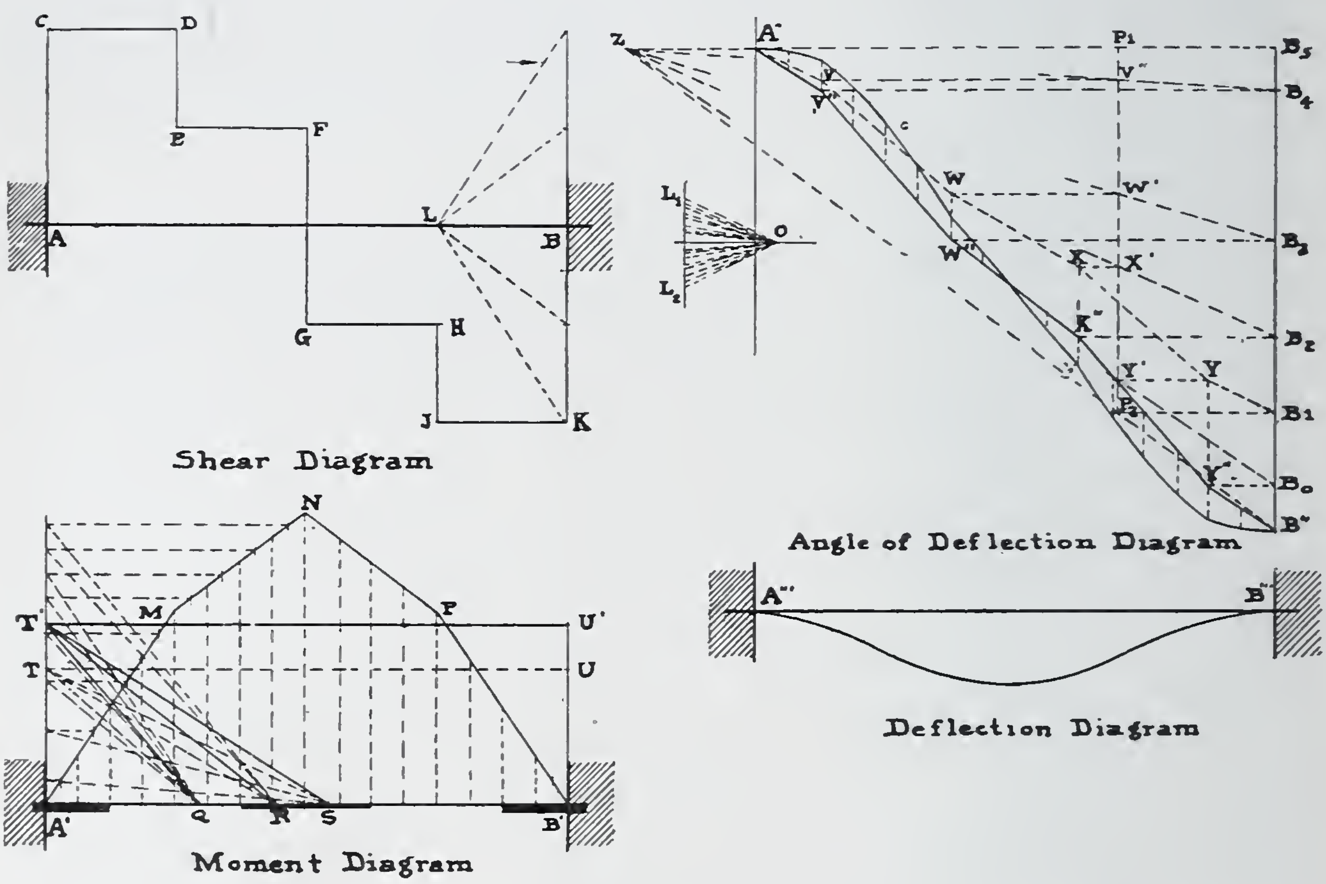
or both of these diagrams. If the position of the mid-point of the line RS is in the right position, the angle diagram will apparnetly check; but the deflection will not check unless the angle of inclination of the line RS is correct. This fact is very important when we come to consider the application of this method to hingeless arches, as it is very necessary to check the moment diagrams before applying them to the arch.

In the case just considered, it might be well to call attention to the fact that the area of the figure A′RSB′ and that of the figure A′NOPB′ must be equal, and a vertical line passing through the center of gravity of one will pass through the center of gravity of the other.

DIAGRAM 12

GRAPHIC INTEGRATION APPLIED TO BEAMS
FIXED BEAM WITH VARYING MOMENT OF INERTIA
SYMMETRICAL LOADING

Variation of *I* indicated by double and triple lines in moment diagram



Passing to Diagram 12, we take up the problem of a fixed beam with a varying moment of inertia. In this case we assume that the variation of the moment of inertia and the loading are symmetrical with the beam.

We draw the shear diagram as for a simple beam, and develop the moment diagram without reference to the beam being fixed or to the variation of the moment of inertia. In the diagram, we have assumed that there are three different values of the moment of inertia. Three poles must then be chosen to correspond to these three values. These are shown by the points Q, R, and S. The angle-of-deflection diagram is then drawn as for a simple beam with a varying moment of inertia, using the pole corresponding to the moment of inertia at each point considered.

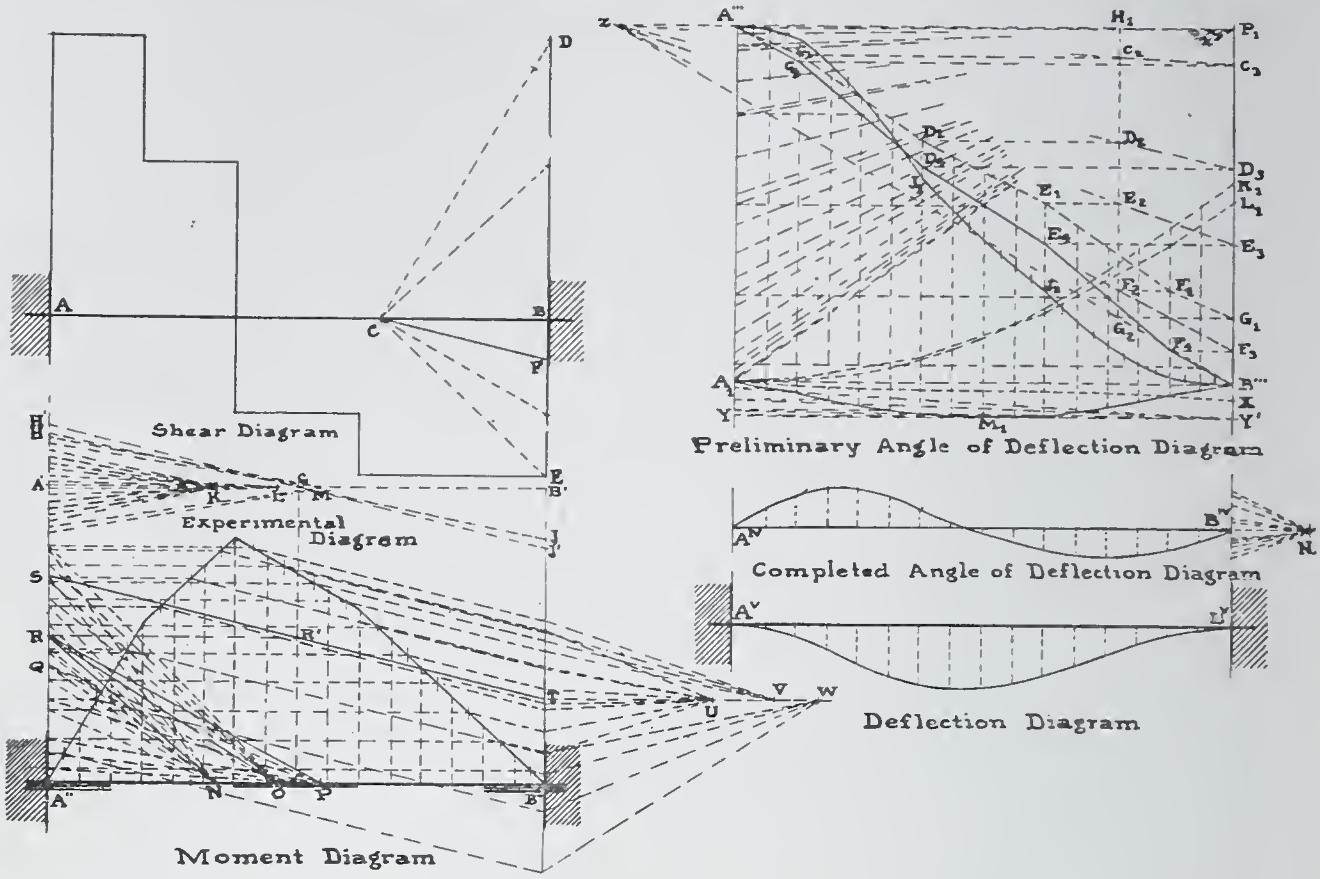
In order to find the true closing line let us assume a closing line, TU, to the moment diagram and drawing the rays ST, RT, and QT, we construct a trial closing line to the angle diagram—$A''VWXY\ B_1$—$A''V$ and YB_1 being parallel to ST; VW and XY being parallel to QT; and WX being parallel to RT. If the assumed position of the closing line, TU, had been correct, then the point B_1 would have coincided with B'', so we see that $A'T$ must be increased in the proportion that B_5B_1 bears to B_5B''. The method shown here is one of several possible methods. It is not so easy as some others, but it has the advantage of giving us a means of checking the accuracy of our work. In this method we choose a point, Z, and draw the line ZB''. Projecting the point B_1 to this line we get the point P_2. We then erect the perpendicular P_2P_1. On this perpendicular we project the points, V to V', W to W', X to X', etc., and from the point Z we draw the lines, $ZV'B_4$, $ZW'B_3$, etc. We then, project the horizontal lines B_4V'', B_3W'', etc., V'' being vertically under V, W'' being vertically under W, etc. We then complete the closing line for the angle diagram $A''V''W''X''Y''B''$. Then drawing ST' parallel to $A''V''$ and $Y''B''$, and the line RT' parallel to $V''W''$ and $X''Y''$, and the line QT' parallel to $W''X''$, we can draw the line $T'U'$, which is the closing line of the moment diagram. If our work has been accurate, the lines from Q, R, and S will meet at the common point T'; if not, it will be necessary to check our previous work.

In order to draw the deflection diagram, we have a choice of two methods, one of which is shown here and the other in Diagram 11. The heights of the angle diagram at the mid-points of the segments can be laid off on the line L_1L_2 and from the pole O the rays drawn and the deflection diagram developed.

DIAGRAM 13

GRAPHIC INTEGRATION OF BEAMS
FIXED BEAM WITH VARYING MOMENT OF INERTIA
UNSYMMETRICAL LOADING

Variation of symmetrical about center of beam. Variation indicated by double and triple lines in moment diagram



In Diagram 13 we have the same case with regard to the beam, but the loading is not symmetrical. Note, however, that the variation of the moment of inertia is considered as symmetrical with the beam. In this case, we draw the shear and moment diagrams as before, and then develop a preliminary angle diagram, $A'''J_1J_2B'''$. An assumed point, Q, is taken and an experimental closing line, $A'''C_1D_1E_1F_1G_1$, is drawn and corrected as before to the line $A'''C_4D_4E_4F_4B'''$. This line gives us the point R instead of Q. We then project the point R horizontally to the center of the girder, giving us the point R′. This is the mid-point of the true closing line. Any line passing through this point if assumed as a closing line would give us a proper integration for the angle diagram; that is, it would show an angle-of-deflection diagram which would apparently close. But, unless the assumed closing line passing through R′ is sloping at the proper angle, the deflection diagram would show a vertical displacement, which is contrary to the requirements of a fixed beam.

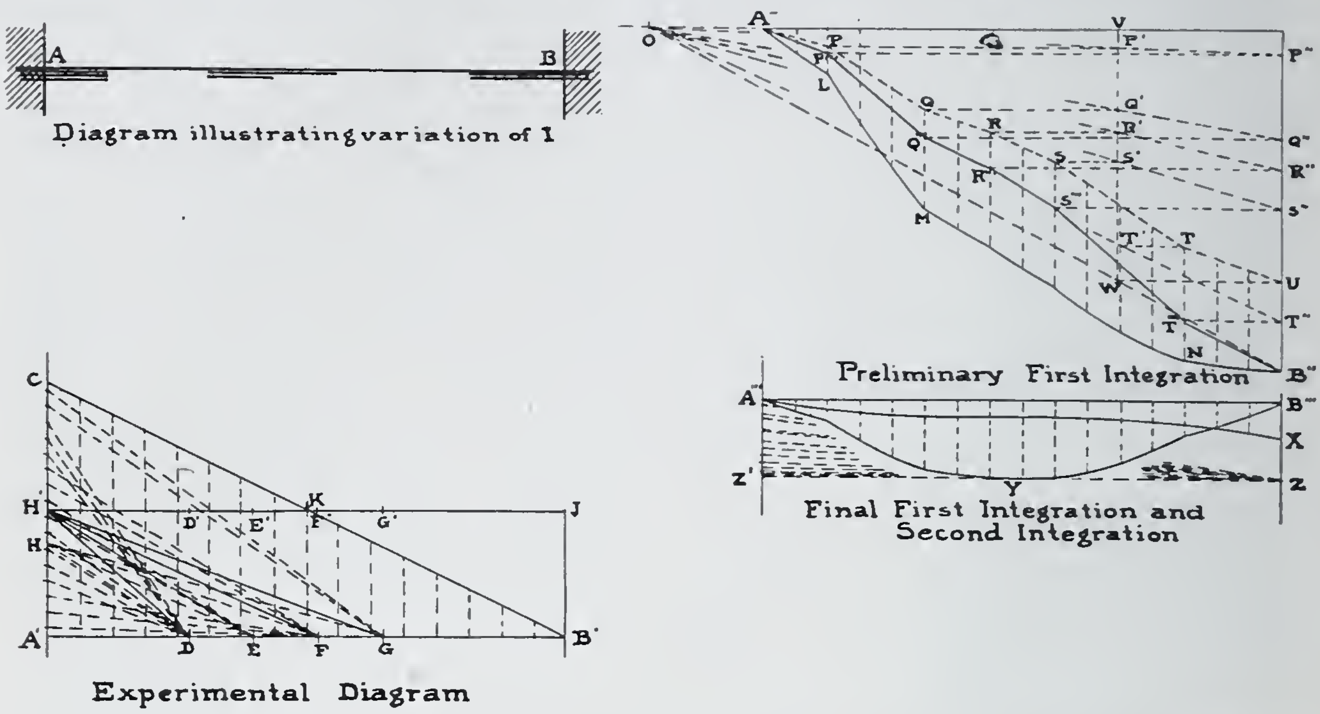
Using P_1 as a pole we integrate the area $A'''P_1B'''J_2J_1$, giving us the line A_1K_1; and we integrate the area $A'''P_1B'''F_4E_4D_4C_4$, giving us the line A_1L_1. As we explained before in connection with Diagram 11, this line K_1L_1 multiplied by $A'''P_1$ represents the difference of the areas of the figures we have just integrated. In order that the deflection diagram may close properly, these areas should be equal. In order to determine the proper angle of slope to give the closing line of the moment diagram, we draw an experimental diagram using $A'B'$ as a base and assuming the line HGJ, which bisects the line $A'B'$ at G. This figure, $A'HGJB'$, is then integrated, using the poles K, L, and M, corresponding to the poles N, O, and P in the moment diagram. We have not shown the division of this figure into segments and the projection of the mid-points. The integration of this figure gives us the figure A_1M_1B'''. Projecting the mid-points of the segments of this figure to the line A_1Y, and using Y' as a pole, we integrate this figure getting the line A_1X. If the assumed slope of the line HGJ had been correct, $B'''X$ would have been equal to K_1L_1. As this is not the case, we lay off $B'J'$ equal to $A'H'$ of such length that $B'J' : B'J :: K_1L_1 : B'''X$. Through R, parallel to $H'GJ'$, we draw $SR'T$, which is the closing line of the moment diagram. Projecting the mid-points of the segments to the line $B''T$ by means of lines parallel to $SR'T$; and on a horizontal line from T, laying off the poles U, V, and W, to correspond to N, O, and P in the moment diagram, we draw the rays and develop the completed angle-of-deflection diagram. Projecting the mid-points of this diagram to the vertical line at the left of the figure and taking N_1 as a pole, we develop the deflection diagram. If our work has been accurate this will close. It will, however, disclose errors very readily. The line CF, as in the case of the beam with uniform moment of inertia, divides the load line into DF—the vertical reaction at the left support—and FE—the vertical reaction at the right support.

DIAGRAM 14

GRAPHIC INTEGRATION OF BEAMS
FIXED BEAM WITH VARYING MOMENT OF INERTIA

Loading and variation of *I* not symmetrical with center of beam

Method same as illustrated in Diagram 13 except development of Experimental Diagram, which alone is shown on this diagram



In Diagram 14 we deal with a fixed beam in which the loading and the variation of the moment of inertia are both unsymmetrical. In this diagram we have omitted everything except the development of the experimental diagram. The process in this case is the same as in the case just considered, except that the experimental line, CB′, will not necessarily intersect the line H′J at the middle point. The assumptions we have made in producing this diagram have been such as to bring the intersection very near to the middle point, but this is a coincidence, due to our having accidentally balanced the variation of the moment of inertia so as to produce this result.

In developing the experimental diagram in this case we must not only determine the proper slope for the closing line of the moment diagram but must also determine the point where it crosses the horizontal line.

To do this we draw for our experimental diagram the figure A′CB′. Using D, E, F, and G as the poles to correspond to the different values of the moments of inertia, we integrate the figure in the usual way. Note that we have not shown the lines projecting the mid-points to the line A′C. By this integration we

obtain the diagram A″LMNB″. Assuming a point, H, and integrating to that point, we get the diagram A″PQRSTU. Correcting this, we get the line A″P‴Q‴R‴S‴T‴B″. Drawing GH′ parallel to A″P‴ and T‴B″, etc., we get the point H′. Drawing the line H′KJ we have our experimental diagram. Using D′, E′, F′, and G′ as poles, we develop in the usual manner the diagram A‴YB‴. The development of this is not shown on this sketch. Using Z as a pole, we integrate the figure A‴B‴Y getting the line B‴X. This corresponds to the line B‴X in Diagram 13; and JB′ and H′C in the experimental diagram must be increased or diminished in the ratio that K_1L_1 (see Diagram 13) bears to B‴X. All the rest of the procedure is the same as explained for Diagram 13.

In all these processes, remember that when the figure we are integrating is bounded by a curved line, the mid-point of the segment we are projecting is to be taken from a point two-thirds of the distance from the center of the chord of the segment to the center of the segment of the curve.

GRAPHIC INTEGRATION APPLIED TO ELASTIC ARCHES

There are many lines through which we might follow the applications of the principles of graphic integration. Among these may be mentioned continuous beams. Another line along which it might be profitable to investigate is that of imperfectly fixed beams, such as beams and girders with web connections at the ends. When we consider how far apart are the calculated breaking stress of such beams, considering them as simple beams, and the actual breaking stress, it seems that there should be some way of determining these stresses. But as we must limit ourselves somewhere, we will consider only the application of these principles to the elastic arch.

To most people, it seems that the determination of the stresses in arches is a problem of an entirely different character from the problem of the determination of stresses in ordinary beams. It is our purpose to show that such is not the case. The parent of all of this tribe of beams and arches is the hingeless elastic arch, with a varying moment of inertia. If the angle of inclination reduces to zero at all points, then the methods and formulas for calculat-

ing the stresses automatically reduce to the methods and formulas for the fixed beam with a varying moment of inertia. If the variation of the moment of inertia reduces to zero, then these methods and formulas again reduce to those for the fixed beam with a uniform moment of inertia. Starting again with our hingeless arch, we find that if the moment of inertia reduces to zero at the ends, then the methods and formulas automatically reduce to those for the two-hinged arch. If the angle of inclination reduces to zero at all points, then we have the simple beam with a varying moment of inertia. If the variation of the moment of inertia reduces to zero, then we have the simple beam with a uniform moment of inertia. The determination of stresses in all of these can be considered as merely different phases of the same problem.

We will first consider the two-hinged arch, since that is the easiest to understand.

In the two-hinged arch, we have the bending moments the same as we would have for a simple beam, but with this exception, that the form of the arch is such that, if unrestrained, part of the deflection would be taken up in lengthening the span, which would be contrary to the theory of the arch—that the position of the supports is fixed. These supports, then, must exert a horizontal thrust, producing a negative moment, which would cause a horizontal contraction equal to the expansion from the moment due to the vertical loading.

There are two conditions, then, which must be met. The first is that the combined moments must not produce any vertical deflection at either end. The second is that there must not be any horizontal movement at either end.

Since the thrust of the abutments to restrain the arch from horizontal displacement is a purely horizontal force, in the symmetrical arch—which is the only kind we are considering in this paper—this force cannot produce any vertical deflection. As the condition that there shall be no vertical deflection at either end is a condition of all beams and arches, except a few special cases which we are not considering here, we conclude that the construction of the moment diagram, in the same manner as for simple beams, will meet this requirement. This is capable of a more elaborate mathematical demonstration but we will not take time

for it as it is simply an application of the principles found in any book on structural mechanics, in the part devoted to deflection of beams.

The second condition—that there shall be no horizontal movement of either end—is met if $\int_0^l \frac{Myds}{EI} = 0$, in which M is the bending moment at any point; y is the height of that point above the springing line of the arch; ds is an infinitesimal segment of the arch rib measured along the neutral axis of the arch; E is the modulus of elasticity; and I is the moment of inertia at the point in question. The proof of this statement is found in any work on arches, and we will not take space to demonstrate it here. The expression $\int_0^l \frac{Myds}{EI}$ is equal to the total horizontal movement of the free end, if one end is free to move in a horizontal direction. All that it is necessary for us to do in the case of the two-hinged arch, is to determine the thrust hereafter denoted by H which, producing a negative moment equal to Hy, will cause a contraction of span equal to the expansion caused by the moment due to the vertical loading (denoted by M). In other words, we must have the following equation:

$$\int_0^l \frac{Myds}{EI} - \int_0^l \frac{Hy^2ds}{EI} = 0.$$

If we let t represent the angle of inclination of the arch at any point, it is evident that $ds = \sec t dx$. If we let $I_x = I \cos t$, the above equation will reduce to the following form:

$$\int_0^l \frac{Mydx}{EI_x} - \int_0^l \frac{Hy^2dx}{EI_x} = 0.$$

If we denote the horizontal displacement by r, then, as above, we will have:

$$\int_0^l \frac{Mydx}{EI_x} = r.$$

If we represent the rise of the neutral axis of the arch by h, then the following will be true:

$$\int_0^l \frac{Mydx}{hEI_x} = \frac{r}{h} \dots\dots\dots\dots\dots\dots (1)$$

And if we represent the horizontal displacement due to the horizontal thrust by r′, then we will have the following:

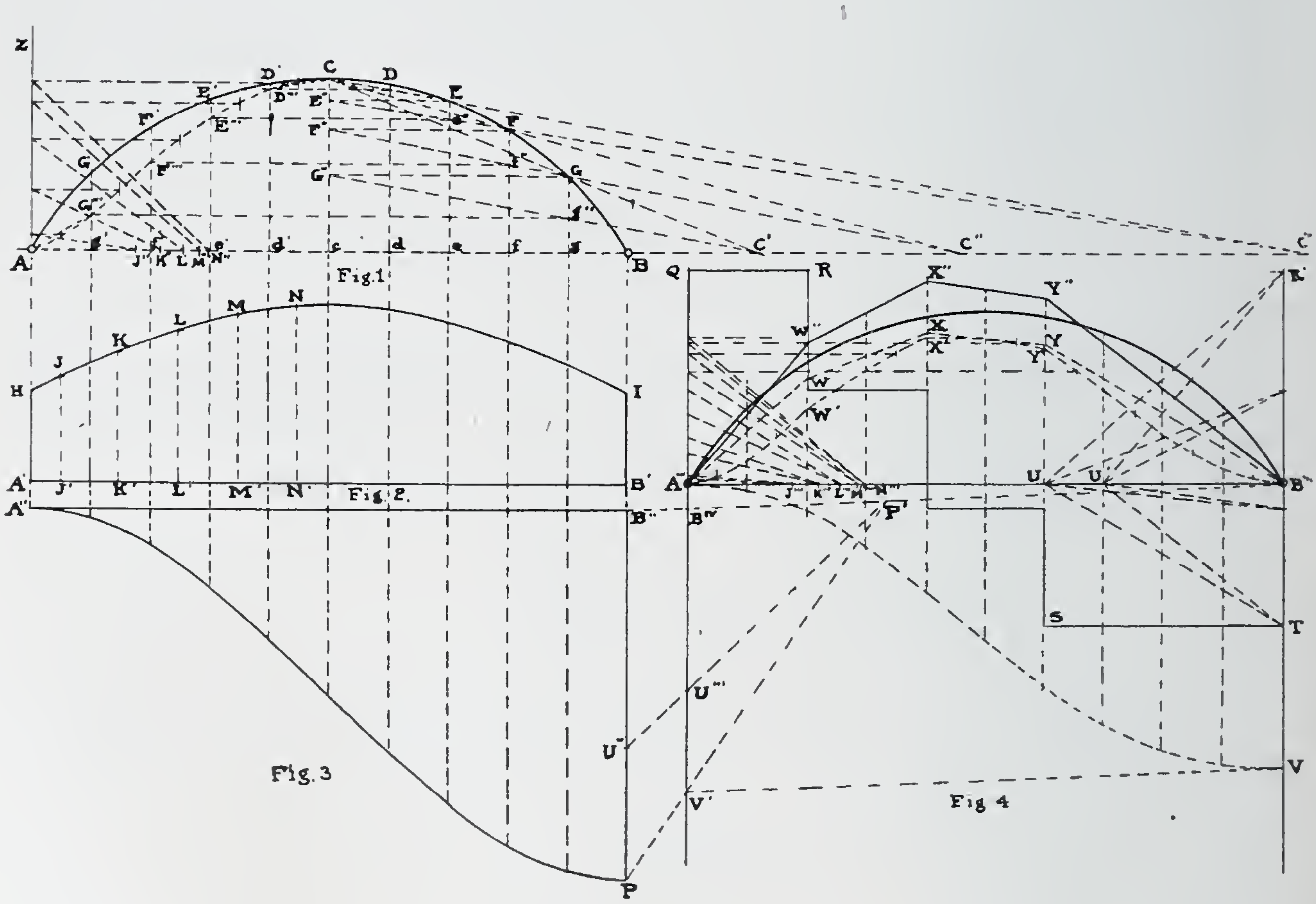
$$\int_0^l \frac{Hy^2dx}{EI_x} = r', \text{ and from this we have:}$$

$$\int_0^l \frac{Hy^2dx}{hEI_x} = \frac{r'}{h} \dots\dots\dots\dots\dots\dots (2)$$

It is on equations (1) and (2) that this method of graphic determination of stresses is based.

DIAGRAM 15

GRAPHIC INTEGRATION APPLIED TO TWO-HINGED ARCHES



Referring now to Diagram 15, we have the two-hinged arch illustrated. This arch may be of any form. The sketch here given is the arc of a circle, but it may be a parabola, part of an ellipse, or even of the Gothic form—the treatment is the same.

In this diagram the neutral axis of the arch is represented by the arc ACB. This is divided into segments, at the points D, E, F, and G on one side, and at the corresponding points D′, E′, F′, and G′ on the other. In Fig. 2 we have the values of I_x, which is equal to $I \cos t$, plotted as ordinates to the figure A′HIB′. In this case we have assumed that the moment of inertia is greater in the center than at the ends of the arch. This is as it should be, for the greatest stresses are at or near the center of the arch. In many of the text-books the assumption is made that the arch is a parabola and that the moment of inertia varies as the secant of the angle of inclination of the arch rib. Why in the world any one, not an idiot, should think of putting the strongest part of the arch rib at the point where the least stress occurs, and the weakest point where the greatest stress occurs, is beyond the writer's powers of comprehension. If it is argued that this assumption is made merely to simplify the presentation of the principles of arch design, why do the authors stop with an utterly impractical example before the student? Why do they not go on and illustrate the application of these principles to a practical case? I once asked an engineer why the authors of the text-books always choose some easy problems to illustrate, and let the hard points go. He answered, "Oh, they have to choose something they can work."

But, to proceed with the solution of our problem, we must construct inside the arch a secondary curve whose ordinates shall bear the same relationship to the ordinates of the arch that the ordinates of the arch do to the height of the arch at the center. In other words, if we represent the ordinates of this secondary curve by y', and the ordinates of the arch by y, then $y' = \frac{y^2}{h}$.

At this point we will explain that hereafter, when we speak of a secondary arch, we are referring to such a curve as the one described, and when we speak of a secondary moment diagram, we mean a diagram in which the ordinates bear the same relation-

ship to the ordinates of the moment diagram that the ordinates of the arch bear to the height of the arch at the center. In other words, if we represent the ordinates of the moment diagram by y_1, and the ordinates of the secondary moment diagram by y_2, then $y_2 = \frac{y_1 y}{h}$.

There are several ways by which this secondary arch may be constructed, each of which has its advantages. One method is shown in this sketch. In this method, lines are drawn from C through G to C′, through F to C″, through E to C‴, etc. Projecting G to G″, we draw the line G″C′ cutting Gg at g″. Then it is evident that gg″ : Gg :: Gg : Cc, and g″ is a point on the secondary arch. For convenience, g″ is projected to the corresponding point in the other half of the arch, and only half of the secondary arch is constructed, as shown by the dotted line AG‴F‴E‴D‴C. Now it is evident that the horizontal thrust in a two-hinged arch can be applied only at the hinges, hence the bending moment at any point due to a thrust, denoted by H, will be equal to Hy. Assuming for convenience that H is equal to unity, we see that the bending moment in such a case would be represented by y or, in other words, to the diagram of the arch. $\frac{My}{h}$ would then be equal to $\frac{y^2}{h}$, and be represented by the secondary moment diagram.

If we indicate by I_j the value of I_x represented by the line JJ′ in Fig. 2 (Diagram 15), and by I_k the value represented by KK′, etc., the scale of the diagram A′HIB′ which we will represent by s, will be equal to $\frac{JJ'}{I_j} = \frac{KK'}{I_k}$, etc.

If we now integrate the secondary arch, we have: $\int_0^l \frac{y^2dx}{h}$.

Projecting the mid-points of the segments of the secondary arch to the line AZ, and laying off the values of the moments of inertia at the mid-points of the segments, as AJ″ = JJ′, AK″ = KK′, AL″ = LL′, etc., and using these points as poles, we integrate the secondary arch as shown in Fig 3 (Diagram 15),

indicated by the line A″P. B″P represents the result of the integration, $(B''P)s = \int_0^l \frac{y^2dx}{hI_x} = \int_0^l \frac{Mydx}{hI_x}$. Then $\frac{(B''P)sh}{E} =$ $\int_0^l \frac{y^2dx}{EI_x} = \int_0^l \frac{Mydx}{EI_x}$, which is the amount of displacement which would be caused by a horizontal pull of one pound acting through the hinges if the ends were free to move.

In order to determine the stresses due to a change in temperature, it is simply necessary to divide the change of length, which would be caused by any assumed range in temperature, by the displacement which would be due to a pull or thrust of one pound. This change, due to a thrust or pull of one pound, is obtained by multiplying B″P, in inches (on the scale of the layout of the arch) by JJ′, also in inches, and that by Cc, in inches, and dividing the product by EI_j.

Bending stresses, due to the shortening of the rib under compression, can be determined in the same way. To get the bending moments due to temperature or rib-shortening stresses, multiply the thrust by y.

To determine the moments due to vertical loading, we proceed as for a simple beam (see Fig. 4). The shear diagram is drawn, A‴QRSTB‴. Assuming a pole, U, we proceed to draw the moment diagram, A‴WXYB‴. We then develop the secondary moment diagram according to the same principles by which the secondary arch was developed. This secondary moment diagram is shown by the line A‴W′X′Y′B‴. Projecting the midpoints of the secondary moment diagram to the vertical line A‴Q, and laying off the poles A‴J‴ = JJ′, A‴K‴ = KK′, etc., we integrate the secondary moment diagram, as shown by the line A‴V. But it is necessary for us to develop a moment diagram which would produce a line, corresponding to B‴V, which would be equal to B″P. As B‴V is less than B″P, it will be necessary for us to reduce the pole of the moment diagram in the same proportion. To do this we draw the line B‴B″. Parallel to this line

we draw the line VV′. $B^{iv}V'$ is equal to B‴V. From P through V′ draw the line PV′P′. Lay off B″U″ equal to UB‴, and draw the line U″P′ cutting the line $B^{iv}V'$ at U‴. Lay off a new pole, U′B‴, equal to $B^{iv}U'''$ and redraw the moment diagram A‴W″X″Y″B‴. As we have now increased the height of the moment diagram in the same ratio that B″P bears to B‴V, and it is evident that if we would develop another secondary moment diagram it would bear the same relation to the one produced from the first moment diagram, it follows that if we were to integrate this new secondary moment diagram, the result would be a line corresponding to B‴V, which would be equal to B″P.

We now draw the neutral axis of the arch from A‴ to B‴. Now we have the line of the arch representing a negative bending moment which would produce a shortening of the span if the ends were free to move, equal to the amount we determined would occur with a one-pound thrust, multiplied by the value of the pole U′B‴. We also have a positive moment diagram, A‴W″X″Y″B‴, which would produce a lengthening of the span by an equal amount. We have then met the second condition of the two-hinged arch—namely, that there shall be no horizontal displacement.

To determine the bending moment at any point, due to vertical loading, we multiply the vertical distance between the arch rib and the line of the moment diagram by the value of the pole, U′B‴, in pounds.

The determination of axial thrusts, etc., will not be discussed here, as it follows so readily that any one familiar with graphics will have no trouble with the matter.

We have explained the steps of this method in detail, but a little experience in handling the method will suggest several short-cuts—for example, the actual construction of the secondary arch will likely be omitted after a few trials of the method, and other such time savers will develop with experience.

HINGELESS ARCHES

In the case of the hingeless arch, we have three conditions to meet. First, there must be no change in the angle of inclination of the arch rib at the ends. Second, there must be no vertical

displacement at either end. Third, there must be no horizontal displacement at either end. The first, which means that the total change of inclination must be equal to zero, is met, when the following equation holds true:

$$\int_0^l \frac{Mds}{EI} = \int_0^l \frac{Mdx}{EI_x} = 0.$$

The second condition is met when the following equation holds true:

$$\int_0^l \frac{Mxds}{EI} = \int_0^l \frac{Mxdx}{EI_x} = 0.$$

The third condition will be met when the following equation holds true:

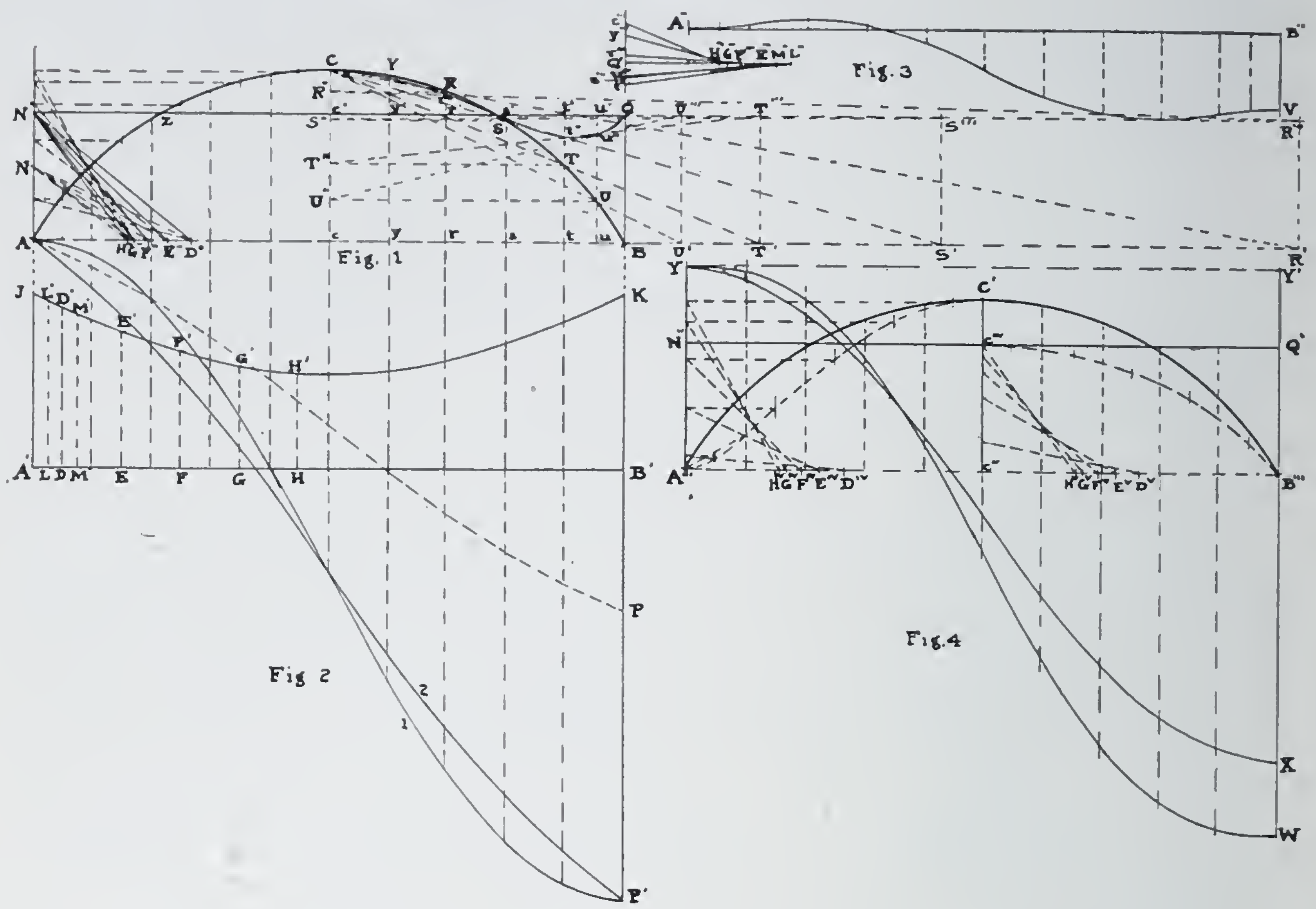
$$\int_0^l \frac{Myds}{EI} = \int_0^l \frac{Mydx}{EI_x} = 0.$$

For proof of these equations see any standard work on elastic arches.

With regard to the second of these conditions, we find, as in the case of the two-hinged arch with a moment diagram for a simple beam, that if we produce a moment diagram as for a fixed beam and check it with the deflection diagram to see that there is no vertical displacement, then we will meet this condition. This is evident when we consider that the arch is simply a beam in which a part of the bending moments are counteracted by a horizontal force applied at the ends. If the bending moment shows no vertical displacement at either end, and the only other force applied is a balanced horizontal force, no vertical displacement can result. There is an apparent but not real fallacy in this statement, when we inspect the moment diagram of a fixed arch with unsymmetrical loading and observe that the points of application of the horizontal forces at the two ends are not at the same elevation. The fact of the matter is that, since the closing line of the bending moment is imposed upon the closing line of the arch, the composite diagram is simply the resultant of two opposing forces, neither of which will produce any vertical deflection.

DIAGRAM 16

GRAPHIC INTEGRATION APPLIED TO HINGELESS ARCHES, 1



Having thus disposed of the second condition, there remain the first and third. Referring to Diagram 16, in Fig. 1 we have the arch ACB. In Fig. 2 we have the moment-of-inertia diagram A′JKB′. As in the case of the two-hinged arch, the ordinates of this figure are proportional to the moments of inertia at the different points, divided by the secant of the angle of inclination of the arch rib at these points. Or, in other words, the ordinates of this figure represent the values of I_x at the different points along the arch. Notice that the values of I_x, in this case, are shown greater at the ends than in the middle. This is in accordance with the usual practice. If the moment of inertia were made to vary as the secant of the angle of inclination of the arch rib, the line JK would be a straight line.

Our first object is to find the closing line of the arch, which should be a line along which any thrust or pull would produce no change of angle at A or B. If a thrust or pull is applied along the line AB, as in the case of the two-hinged arch, then we would have the line ACB of the arch rib representing the moment diagram for such a force. Calling this moment a negative moment.

it is necessary for us to find the height above the line AB at which we may apply the horizontal force so as to produce a positive moment which would exactly counteract the change of angle at A and B. As the change of inclination is represented by the formula $\int_0^l \frac{Mdx}{EI_x}$, we will integrate the negative diagram ACB. Laying off AH″ = HH′, AG″ = GG′, etc.; using these points as poles for the segments they represent; and integrating; we get the line AP′, marked 1. Now, assuming a point, N, and drawing rays from it to the different poles H″, G″, etc., and drawing the integrating line, we get the line AP. But in order that there shall be no change of angle at A and B it is necessary that the point P should coincide with P′. The resultant of the integration, BP, should be equal to BP′. In order to accomplish this we must increase the value of AN in the ratio that BP bears to BP′. We lay off AN′ of such length that AN : AN′ :: BP : BP′. Drawing new rays and integrating again, we get the line AP′, marked 2. This second integration is not really necessary, but is useful as a check on the accuracy of our work. Drawing the line N′Q, we have the closing line of the arch, which is the line along which the thrusts or pulls due to changes of temperature are considered as acting. A thrust or pull along this line will meet the first condition, and also the second.

Our moment diagram, then, is represented by the figure AN′Zc′QBSCZA. We may construct a secondary diagram Qu″t″r″C. The method shown here is the same as previously explained. We pass a line through C and U to U′, project U to U″, and U′ to U‴, on the extension of the closing line of the arch. Then we draw the line U‴U″ and the point where this line cuts Uu′, marked u″, is a point in the secondary diagram. The mid-points of the segments of this secondary diagram are laid off on the line t‴Q′c‴—the closing line being represented by Q′. Laying off the values of I_x we integrate as in Fig. 3, getting the line A″V. Another way of doing this is to draw a secondary arch, as shown by the dotted line A‴C′ in Fig. 4, and the secondary closing-line figure shown by the dotted line B‴c‴. Only half of each of these secondary figures is shown. The sec-

ondary arch can then be integrated, giving us the line YW. The secondary closing-line figure can also be integrated, giving us the line YX. If our work has been accurate, WX will be equal to B″V. Note that the secondary closing-line diagram is proportional to the arch at all points and it is not necessary to integrate this figure, as we may simply lay off Y′X of such a length that Y′X : BP′ :: AN′ : Cc ($= h$).

If we now use the same system of notation used with the two-hinged arch,—that is, I_h representing the moment of inertia corresponding to HH′, I_g representing that corresponding to GG′, etc., and s representing the scale to which the figure A′JKB′ is laid out—that is, $s = \frac{HH'}{I_h} = \frac{GG'}{I_g} = \frac{FF'}{I_f}$, etc.—then if we let M indicate the moment caused by a thrust or pull along the line N′Q; let M' indicate the moments represented by the figure of the arch, ACB; and let M'' indicate the moments represented by the figure AN′QB; and remember that B″V is equal to WX, we will have the following equations:

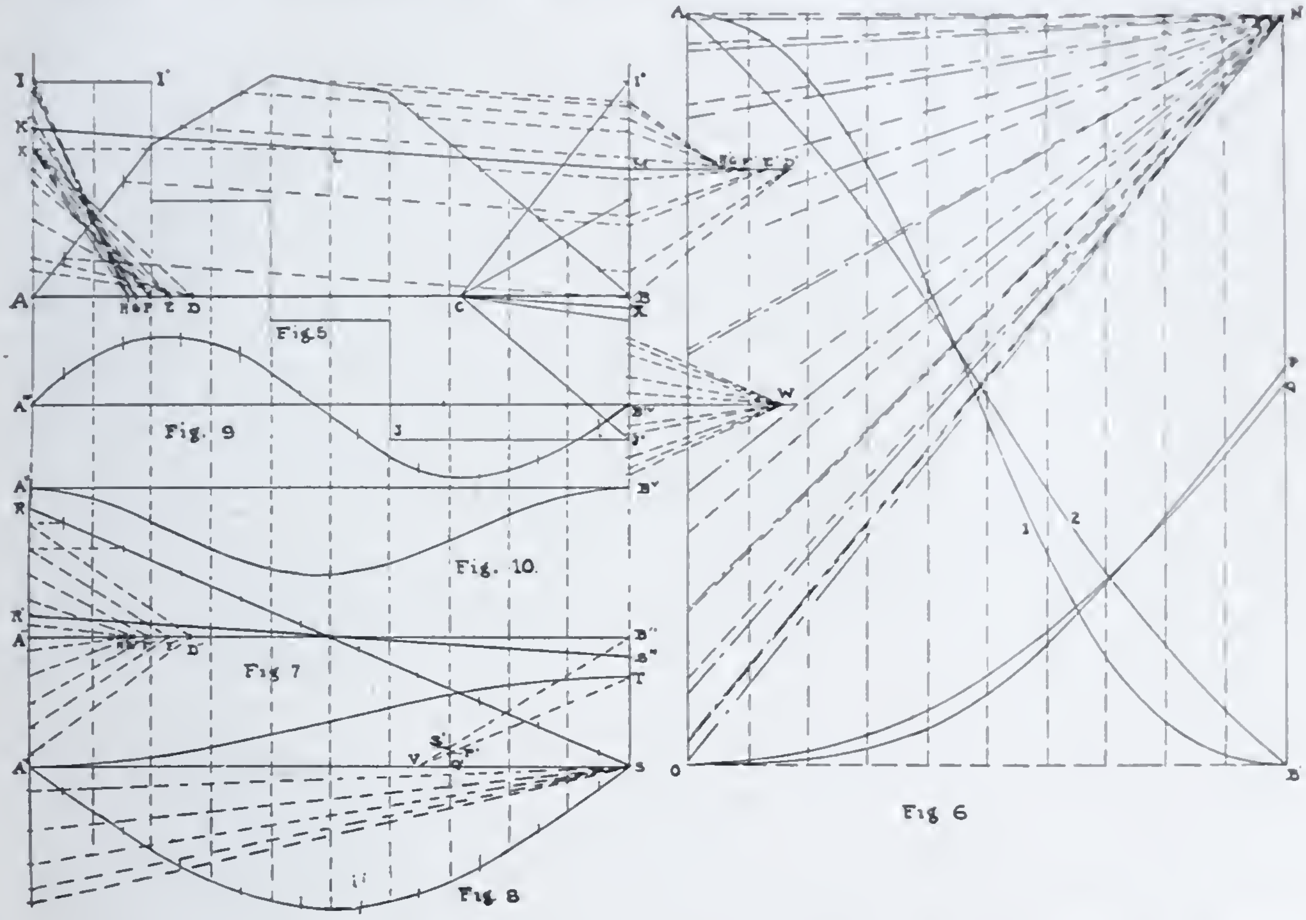
$$(B''V)s = \int_0^l \frac{Mydx}{hI_x} = \int_0^l \frac{M'ydx}{hI_x} - \int_0^l \frac{M''ydx}{hI_x}$$

$$\text{Then } \frac{(B''V)sh}{E} = \int_0^l \frac{Mydx}{EI_x}.$$

To determine the horizontal displacement, which would be caused by a thrust or pull of one pound along the line N′Q, we multiply B″V (measured in inches by the scale to which the arch is laid out) by HH′ (in inches to the same scale) and the product by Cc ($= h$) (also in inches) and divide the product by EI_h. We could take any other ordinate of the figure A′JKB′ if we wished but, in such a case, we should use the corresponding moment of inertia in the divisor. Remember that the values of I_h, I_g, etc., which we use, must represent the moment of inertia divided by the secant (or multiplied by the cosine) of the angle of inclination of the arch rib at the point in question.

DIAGRAM 17

GRAPHIC INTEGRATION APPLIED TO HINGELESS ARCHES, 2



Leaving Diagram 16, for the present, and passing on to Diagram 17, we proceed to the determination of the stresses due to vertical loading. This diagram shows the development of the moment diagram for a fixed beam with a varying moment of inertia, the variation of the moment being symmetrical. This process has been fully explained as illustrated in Diagram 13.

In Diagram 16, Fig. 5 shows the shear diagram AII′JJ′ from which, using the point C as a pole, the moment diagram as for a simple beam is developed. Transferring the values of I_x (AH, AG, AF, etc.) from Diagram 16, we project the mid-points of the moment diagram to the line AI, and drawing the rays, we integrate, producing the line A′B′, marked 1. In this diagram we have omitted the experimental point, corresponding to point Q in Diagram 13, and show only the point K, corresponding to the point R in Diagram 13. This point, K, is projected over to L at the center of the arch, which is the center point of the closing line of the moment diagram. Drawing the rays from H, G, F, etc., we integrate, getting the line A′B′, marked 2, in Fig. 6. Projecting the mid-points of the lines 1 and 2 in Fig. 6 and using

the point N as a pole, we integrate the area of A′NB′, as bounded by line 1, getting the line OP. We also integrate the figure as bounded by the line 2, and get the line OQ. PQ multiplied by A′N is the difference between the areas of the figure A′NB′ as bounded by the two lines 1 and 2. Fig. 7 shows the experimental diagram. Drawing the line RS, and the horizontal line A″B″ intersecting RS at the center, and laying off the values of I_x, (A″H″, A″G″, etc.) to correspond to AH, AG, etc., in Fig. 5, and drawing the rays, we integrate the figure A″RSB″. This gives us the figure A‴US. Projecting the mid-points of the segments of the line A‴US and using S as a pole, corresponding to N in Fig. 6, we integrate this figure and get the line A‴T. To get the true slope of the closing line of the moment diagram we must reduce the slope of the line RS to a slope which will after the two integrations give us a line, corresponding to ST, which will be equal to PQ. To correct this slope, we lay off at a convenient distance from S the line P′Q′, equal to PQ. Through the point P′ we pass the line TP′V. We then draw the line B″V, intersecting the line extended from P′Q′ at S′. Then we have the proportion, TS : P′Q′ :: B″S : S′Q′. We then lay off B″S″ and R′A″, each equal to S′Q′, and draw the line R′S″. Then in Fig. 5, we draw, through the point L, the line K′LM parallel to R′S″. This is the closing line of the moment diagram. We then draw the line CX, parallel to K′LM, which gives us the vertical reactions at the ends—I″X at the left end and XJ′ at the right.

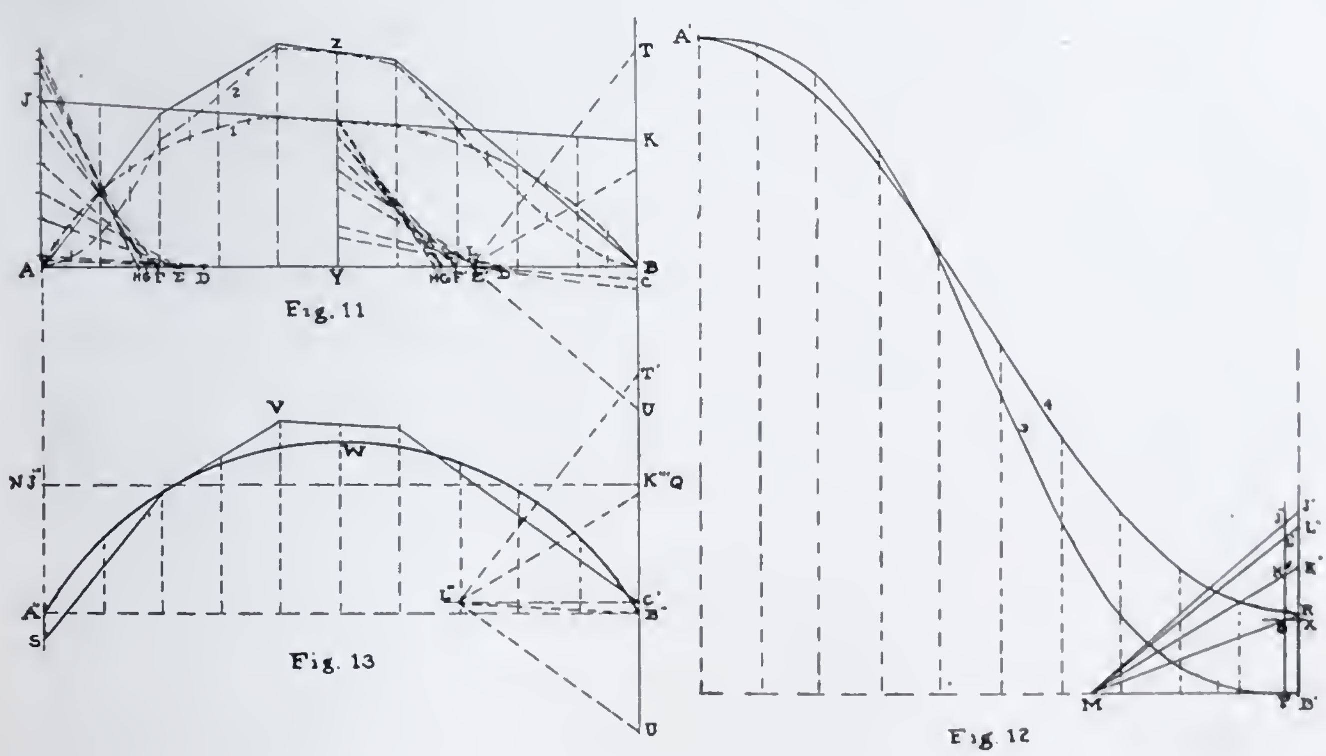
We can now project the mid-points of the segments—by lines parallel to the line K′LM—to the vertical line BI″; and, laying off the values of I_x on a horizontal line from M, we can develop the angle-of-deflection diagram, Fig. 9. Projecting the mid-points of the segments of this diagram to the vertical line at the left; choosing a pole, W, on the line $A^{iv}B^{iv}$ extended; and drawing the rays, we can develop the deflection diagram, Fig. 10. These last two steps—the development of the angle and deflection diagrams—while not necessary to the solution of the problem, are very important as a check on the accuracy of our work.

Just to illustrate the importance of this accuracy, we would call attention to the fact that in so many of the processes, in deal-

ing with fixed beams and hingeless arches, we are dealing with differences in values instead of the values themselves—as, for instance, the line PQ in Fig. 6. This is the difference between B′P and B′Q. Now it is commonly considered that an error of, say, three per cent. in a graphic process is not of serious import. But, supposing that we allow that in this operation—letting the value B′Q be represented by 100, and B′P by 110—then the value we are after is 10. If an error of three per cent. is allowed, and the operator gets the value 103 for B′Q and 107 for B′P, his error is not three per cent. but 60 per cent. But the development of the angle and deflection diagrams will give us a very close check on our work. The operator must be honest with himself, however, and correct his work if any error shows up, and not fudge his lines to make them close.

DIAGRAM 18

GRAPHIC INTEGRATION APPLIED TO HINGELESS ARCHES, 3



Passing on to Diagram 18, we have in Fig. 11 a repetition of the moment diagram as developed in Diagram 17. In the load line at the left of this figure, BT corresponds to BI″ in Diagram

17 ; BU to BJ′; point C to point X. The dotted line marked 2 represents the secondary diagram of positive moments, while the dotted line marked 1 represents the secondary diagram of negative moments. The mid-points of the segments of the line 2 are projected to the line AJ, and those of line 1 to the line YZ. Laying off our values of I_x, as before, we integrate line 2, getting the line A′B′, marked 3, in Fig. 12. We also integrate the line 1, getting the line A′R. Then RB′ bears the same relationship to the diagram (Fig. 11) that the line B″V, in Diagram 16 does to the arch diagram.

To get our final diagram (Fig. 12) we must reduce the diagram in Fig. 11 till the line corresponding to RB′ is equal to the line B″V in Diagram 16. To do this, we select a point M, at a convenient distance from B′, and draw the line RM. The line B′X is laid off equal to B″V (Diagram 16) and the point X projected horizontally to where it will intersect MR at O. We then erect the perpendicular PO and extend it as far as necessary. Now, to reduce the moment diagram we must increase the pole; so PL′ is laid off equal to BL, and the line ML′L″ drawn. The new pole for the moment diagram is B′L″. Since BK and AJ must be reduced, B′J′ equal to AJ, and B′K′ equal to BK are laid off and the lines J′M and K′M are drawn. This gives us the revised ordinates for the closing line. Laying off our load line again, at T′C′U′ (Fig. 13) and this time placing our new pole opposite the point C′, corresponding to C in Fig. 11; in order that the closing line of the diagram may be horizontal, we draw the line J‴K‴ for the closing line, making K‴C′ equal to PK″. For convenience, we use C′ as a starting point for the new moment diagram, which will finish at S. If our work has been accurate, J‴S will be equal to PJ″. Now, we go back to Diagram 16, and take the arch diagram and placing the closing line, N′Q, on the line J‴K‴ in Diagram 18, we draw in the arch A″B″. The fact that one of the load points comes at B″ is a coincidence and the line L‴B″ has nothing to do with the arch diagram.

We have now imposed the closing lines of two diagrams—one for the vertical loads and the other for the horizontal thrust—upon each other. Each of these diagrams meets the first condition—that there shall be no change of angle of inclination of the arch rib at either end. Each of them also meets the second condition—that there shall be no vertical displacement at either end. Each of these diagrams will show the same horizontal displacement, but in opposite directions. We have then met the three conditions of the hingeless elastic arch, and the diagram A″WB″C′VS is the moment diagram for the arch under the vertical loading assumed.

To determine the moment at any point, multiply the vertical distance between the moment diagram for the vertical loads and the line of the arch rib, by the pole L‴C′ in pounds.

To get the temperature stresses we simply divide the change of length, which would be caused by the change in temperature, by the change due to a pull of one pound along the closing line—explained in our discussion of Diagram 16. Effects of axial compression may be calculated in the same manner.

One point might be brought out here—that, if the moment of inertia of the arch rib varies as the secant of the angle of inclination, the vertical reactions for any given loading will be the same as for a fixed beam with a uniform moment of inertia under the same loading. Also, if we had a fixed beam whose moments of inertia if plotted as ordinates would give the same curve as is given by plotting the values of I_x ($= I \cos t$) for the arch—as shown by the figure A′JKB′ in Diagram 16—then the vertical reactions at the ends, for any given loading, will be the same for the arch and beam irrespective of the form of the arch.

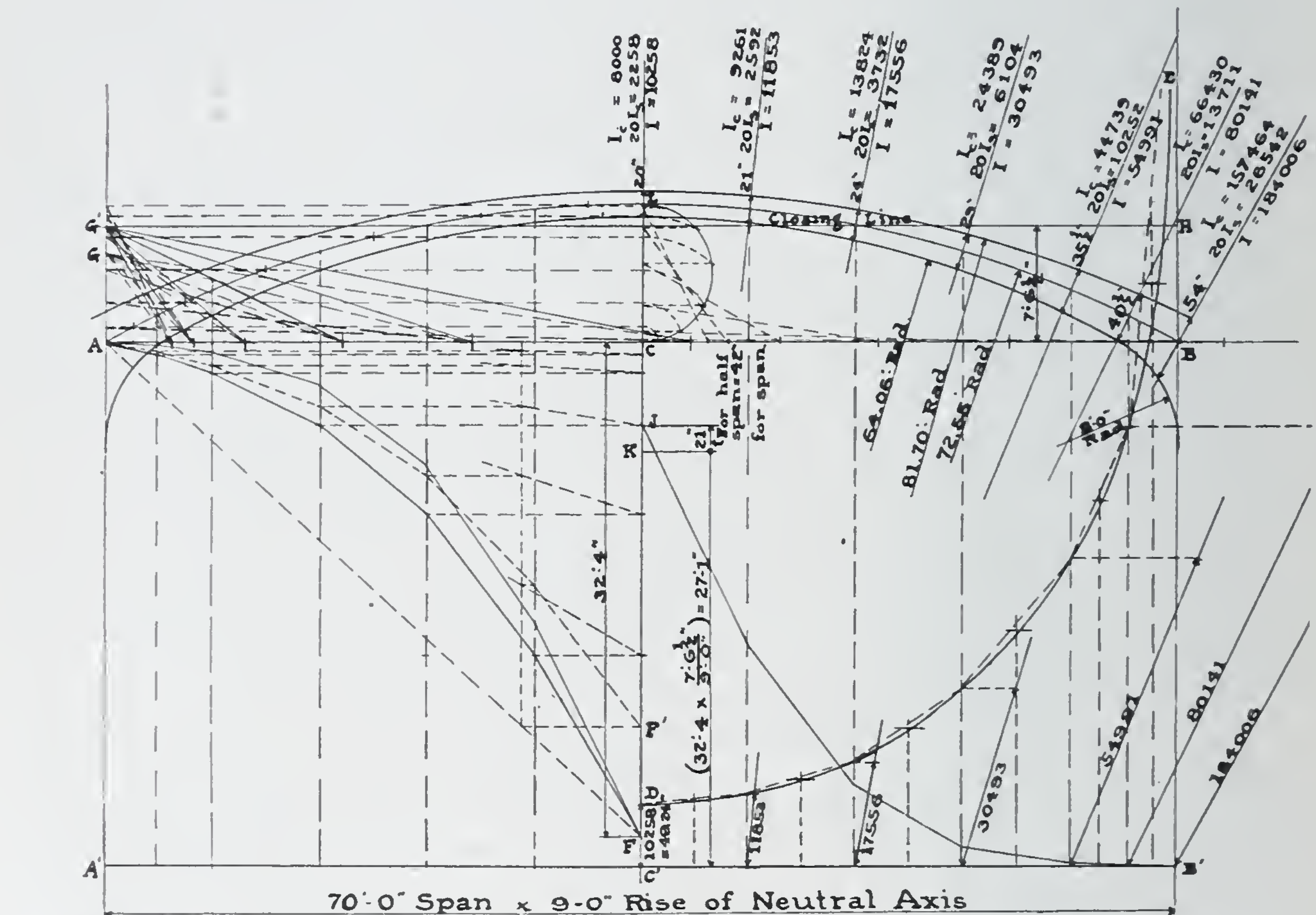
Two questions naturally suggest themselves to us. First, how accurate is the method? and second, how rapid is it? To the first, we would answer that it depends on the operator; a careful operator, with a good eye and a sharp pencil, ought to produce by this method results more accurate than with any other of the approximate methods. With a careless operator, however, this method can produce some very erratic results. A weakness will develop in this method when the neutral axis of the arch becomes vertical at the ends. This condition is not often met in

practice. To the second question we would answer that the first time it is tried the method seems tedious, but on the second or third trial it becomes much easier and the writer feels confident in saying that it will likely prove to be the easiest method yet devised for general application.

For purposes of comparison the writer selected an arch in a text-book, worked out by Professor William Cain's method. The writer does not assume any responsibility for the assumptions in working out this arch, as he has simply followed the bases of calculation taken by the author of the text-book. Where there has been any disagreement in results the writer has taken the trouble to calculate the values, with the result that this method seems to be more accurate than Professor Cain's method, at least so far as applied by the author of the text-book.

DIAGRAM 19

PRACTICAL ILLUSTRATION 1



Change of length due to one pound pressure along closing line $= \frac{2 JK \times C'D \times CL}{E_c\, I_{ctr}} = \frac{42" \times 49.24" \times 108"}{1{,}500{,}000 \times 10258} = .0000145152"$

Change of length due to 40° change in temperature $= AB \times e \times t = 840" \times .000006 \times 40° = .2016"$

Thrust due to 40° change in temperature $= .2016" \div .0000145152" = 13{,}890$ lbs. (Thrust by calculation = 13,650 lbs.)

Diagram 19 gives the dimensions of the arch. The position of the closing line is found by the diagram AF, only half of this being used. The secondary arch is not drawn. Instead, the mid-

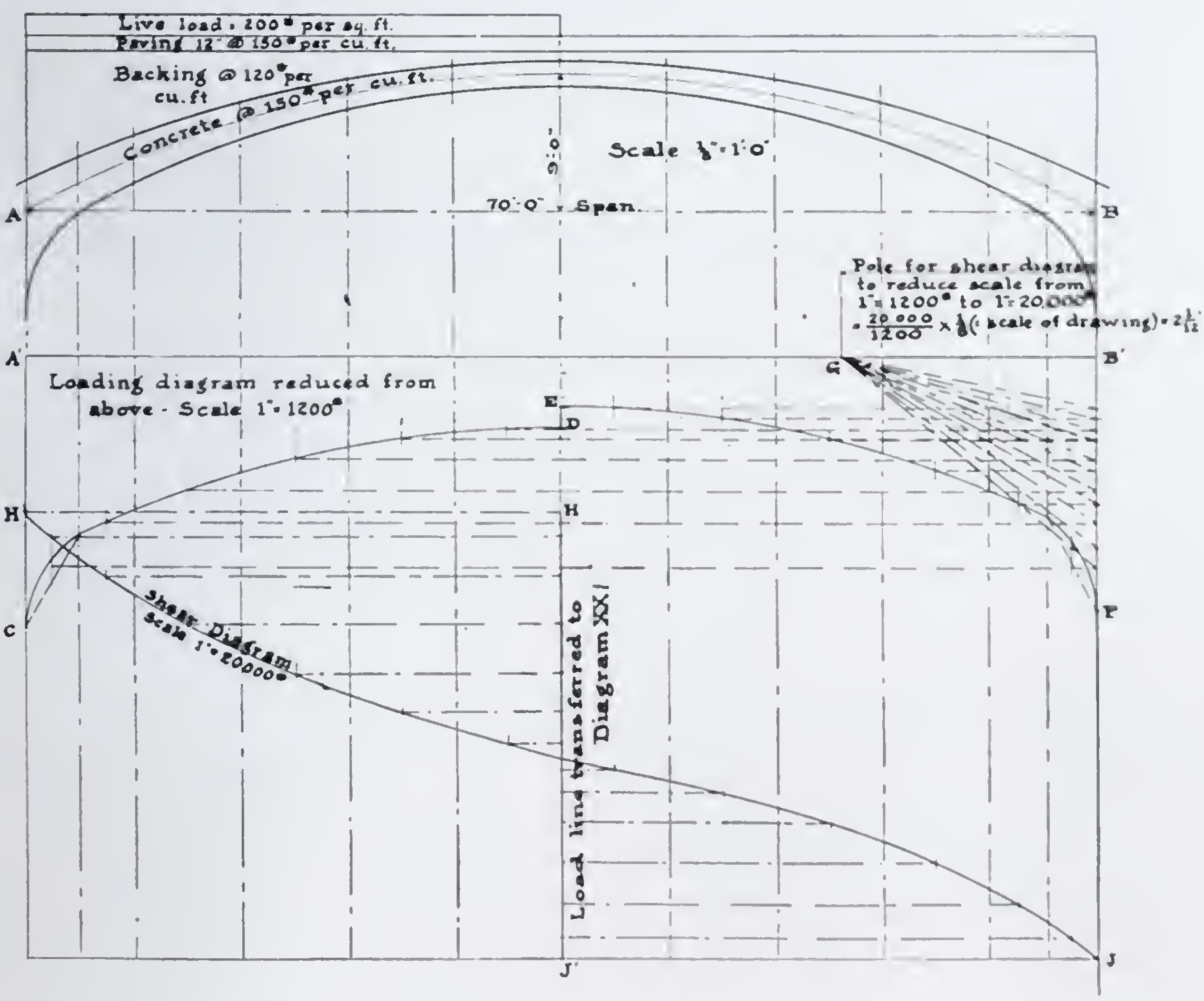
points of the segments of the arch are projected to the center-line of the arch, and a semicircle drawn on LC. With C as a center, we describe arcs from the points where these projections fall on the line LC to where the arcs will cut the semicircle. From these points, lines are projected over to the line LC, which will give us the projections of the mid-points of the segments of the secondary arch.

Note that the values of I_x are taken at points two-thirds of the distance from the chord to the curve DE. The integration of the secondary arch, B′J, is carried out for only half the span. The integration of the secondary figure to AG′HB is simply done by reducing the value of CF in the ratio of the height of the closing line, to the rise of the neutral axis of the arch, giving us C′K.

The results obtained in this case were obtained from this sketch, which was made to the scale of one-eighth of an inch to

DIAGRAM 20

PRACTICAL ILLUSTRATION 2



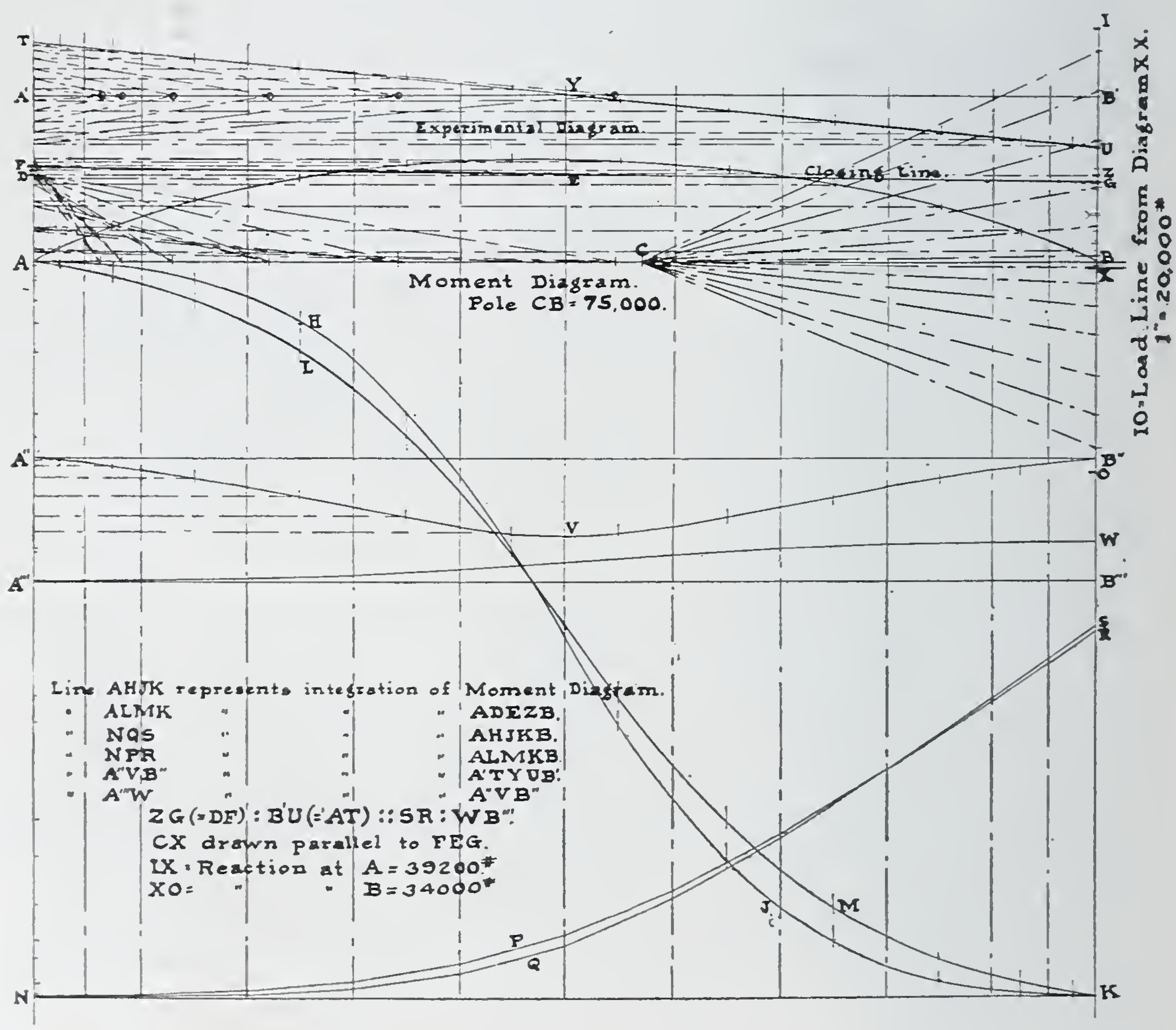
one foot. Note how closely the results as to temperature stresses check with the calculated stresses.

The author of the text-book obtained 8680 pounds as the temperature thrust on a section of the rib one foot wide, under the same conditions. The author of the text-book, must have made some mistake, as Professor Cain's method is certainly more accurate than that. The calculated height of the closing line is 90.04 inches. On the sketch it is given as 90.5 inches, while the author of the text-book gets 92 inches.

Diagram 20 shows the reduction of the load line. The loading given is the same as given in the text-book. Below, we reduce the volume of the backing to give us an equivalent of 150 pounds per cubic foot. The loading diagram is then integrated to find the shear diagram, and the load line, J'H', transferred to Diagram 21.

DIAGRAM 21

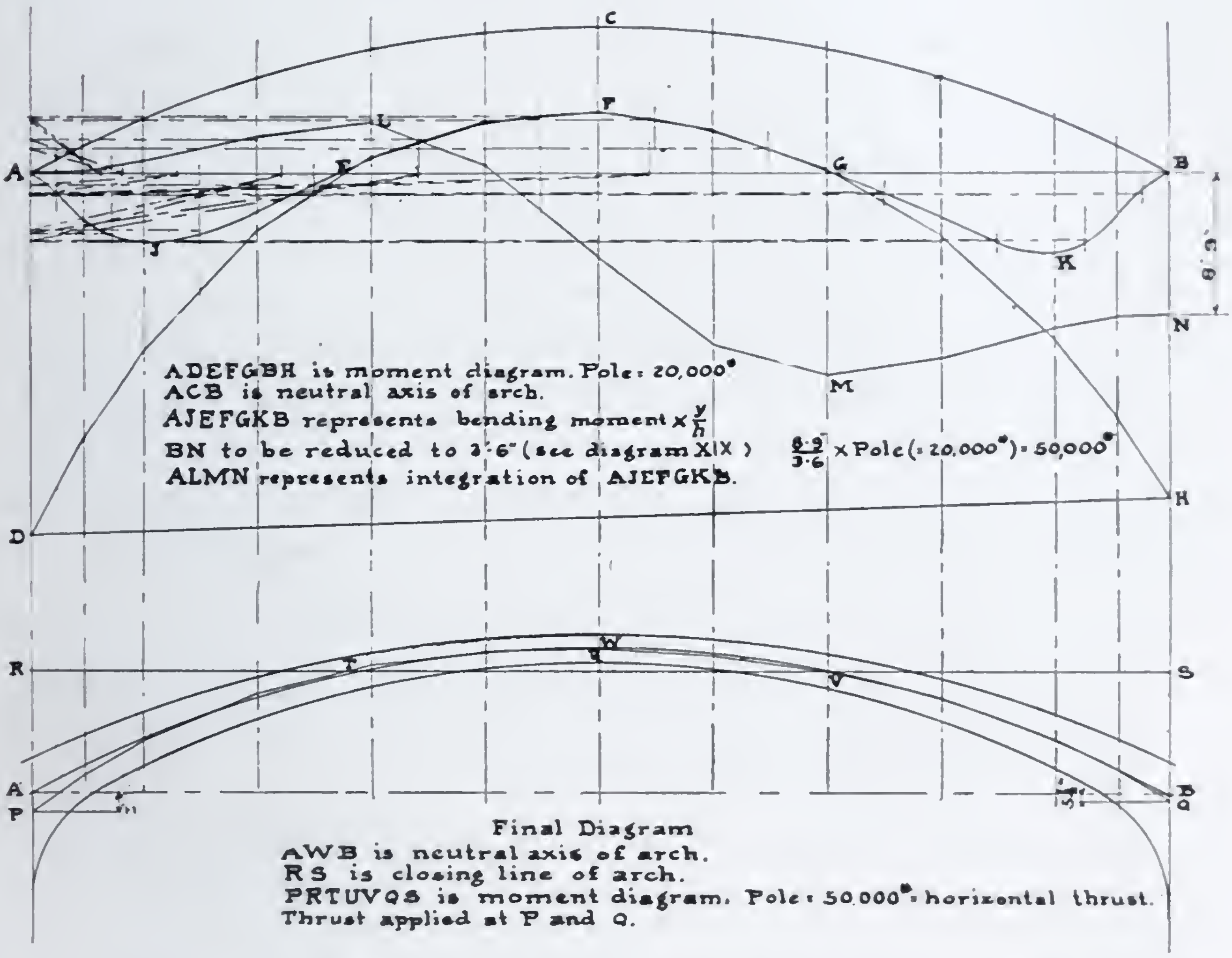
PRACTICAL ILLUSTRATION 3



In Diagram 21, we develop the moment diagram by the methods already described, which we will not repeat.

DIAGRAM 22

PRACTICAL ILLUSTRATION 4



In Diagram 22, the moment diagram is redrawn to a more convenient scale. The reason that the moment diagram was drawn to so small a scale in Diagram 21, is that we wished to illustrate the development of the diagram within the limits of the size of the sheet. In practice, this redrawing would not be necessary.

The secondary diagram, AJEGKB, is drawn and integrated using the values of I_x as shown on Diagram 19, thus developing the line ALMN.

BN must then be reduced to the value of 2(JK) (Diagram 19). This is worked out on the sketch, giving us a pole of 50 000 pounds, which is the horizontal thrust. The final diagram is then drawn by the methods described.

In conclusion, it would seem to the writer that it would be a more scientific way than the usual method of designing the arch, if the ratio of variation of the moments of inertia were first determined and the moment diagrams developed. Then, when we have the stresses practically settled, determine the thickness of the arch rib. This seems to be better than to settle on a thickness, and then work the stresses all out to see if it is strong enough.

NOTATION FOR TABLES SHOWING ECONOMICAL DEPTHS FOR PLATE-GIRDERS

Class 1. Girder of uniform section throughout.

Class 2. Girder with cover plates for parts of its length in following arrangement: One-half of flange section continuous for full length of girder; one-fourth extending for three-fourths of the length; one-fourth extending for half the length.

Class 3. Girders with partial length cover plates as follows: Five-eighths of flange section continuous; three-sixteenths extending for five-eighths of length; three-sixteenths extending for seven-sixteenths of length.

Class 4. Girder with three-fourths of flange section continuous and one-fourth extending for one-half the length.

Division I. Web considered as taking shear only.

Division II. One-eighth of sectional area of web considered as acting with each flange.

Division III. One-sixth of sectional area of web considered as acting with each flange.

M = Bending moment in inch-pounds.

f = Allowed fiber stress in flanges.

t = Thickness of web plates.

t' = Equivalent additional thickness of web plates if total sectional area of stiffeners and fillers is considered as spread over entire surface of web—e.g., if total sectional area of stiffeners and fibers is 30 square inches and length of girder is 20 feet (240 inches),

then

$$t' = \frac{30 \text{ square inches}}{240 \text{ inches}} = \tfrac{1}{8} \text{ inch.}$$

Ordinarily, t' will average from 0.125 to 0.2 inch.

$n = \dfrac{t + t'}{t}$, hence $nt = t + t'$.

y = Depth of girder between centers of gravity of flanges.

S = Shear in pounds.

a = Net area of each flange in Class 1.

a_m = Maximum net area of each flange in Classes 2, 3 and 4.

A = Total net sectional area of girder in Class 1.

A_a = Average total net sectional area of girders in Classes 2, 3 and 4.

A_x = Average net sectional area of flanges and web only, in girders in which stiffeners and fillers are considered.

DISCUSSION

Mr. John A. McEwen:* I was just thinking, while Mr. Martin was speaking, of the old saying "In time of peace prepare for war." We are just reversing that. In time of war he is preparing for peace; which I believe is also a good policy, as we should more than ever now give our attention to economy and conservation.

Mr. Martin has evidently worked out and demonstrated his theories with an infinite amount of patience and labor. During the last few years the method used in designing girders and beams was to look up your stock list and make your design to suit the stock on hand rather than on the basis of strict economy. This was made necessary by the fact that only a limited variety of sections was available, and a design based on the most economical lines frequently meant that you were unable to secure deliveries at all.

The study and development of these theories is an excellent mental drill, but most of us who design do not have the time and could not afford to exhaust our energies going into such matters in great detail.

Personally, I have gotten away from most of my mathematics, the business end of the contracting game taking most of my attention. Designs in our office are made on short-cut methods, and it would probably astonish a professional mathematician to see how quickly we can design a building. I would also add that these designs are very practical and very efficient. As to discussing the theories developed in Mr. Martin's remarkable paper, I would not attempt it at this time.

Prof. Horace R. Thayer:† This paper constitutes an exceedingly valuable contribution to a most fascinating technical

*Pittsburgh Bridge & Iron works, Pittsburgh.

†Professor of Structural Design, Carnegie Institute of Technology, Pittsburgh.

subject. Some of the methods are quite new to me and I would be interested in learning where they originated. It seems also that proofs of the construction employed would be desirable.

The speaker's language rather conveyed the impression that I had originated the method employed in the "Design of Simple Structures" for finding the deflection of plate-girders. While not often employed on this side of the water, this solution is generally attributed to Mohr as is indicated in the above mentioned reference.

I have for several years employed for plate-girders the formula:

$$h = 1.75 \sqrt{\frac{M}{s}}, \text{ where}$$

h = economical depth, back to back of angles;
M = maximum bending moment;
s = allowable flexural stress;

all units being in inches and pounds. It will be noted that this agrees roughly with the speaker's formulas.

I do not favor great accuracy in work of this kind, for the total weight varies slowly in the vicinity of the minimum. Also, small and apparently insignificant clauses in the specification often make a great deal of difference in the true economic depth.

I agree with the preceding speaker that practical conditions may often largely eliminate technical efficiency. War conditions demand speed, and then more speed, let the cost and the weight be what they may. But the return of peace will again bring the vital question of economy to the front. Indeed, it appears that we will face the keenest of competition in our endeavor to retain a portion of the foreign trade on which will depend a large portion of our future prosperity; and our success in that field will depend, in turn, upon the technical efficiency of the engineering profession.

MR. GEORGE W. NICHOLS:* The paper as presented was very interesting and instructive. Whenever Mr. Martin has a paper we can expect something good, and I am sure we were not

*S. C. Webb Engineering Co., Pittsburgh.

disappointed in the paper of the evening. It will be valuable for our PROCEEDINGS on account of the originality of the methods used. As stated by Mr. McEwen our work for the past year or two has been spent mostly in locating material available for girders and other steel-work, with little thought of economy in the sections used. Now that war is over, business will become normal, competition will be keener and it will be well to avail ourselves of the data here furnished.

I think that Mr. Martin should be congratulated on his carefully prepared paper.

AUTHOR'S CLOSURE: In answer to Professor Thayer's inquiry as to where the methods disclosed in this paper originated, the writer would say that wherever he has discovered that other writers have preceded him in the development of any of these methods he has endeavored to give credit to these authors. If there is any author to whom proper credit has not been given, it is because his work has never come to the writer's attention.

As to the proofs of these methods, the formulas for economical depth of plate-girders are developed by the process of differentiating the formula, for the total sectional area of any given condition, in which formulas the depth is the variable. Then, assuming this differential as equal to zero; the value of y, representing the depth, is easily determined.

As to the method of graphically determining the rivet pitch in flanges of plate-girders, in the case shown in Diagram 1, where there is no vertical shear in the rivets, the proof is so obvious as to make its presentation unnecessary.

As to the other methods of determination of rivet pitch in flanges of plate-girders, in cases where there is a vertical component to the shear in the rivets, the writer would say that in his opinion the presentation of these proofs, while not difficult, would have made the paper tedious and, as the writer expects to publish a more extended treatise on these methods in the near future, he did not think it necessary to present all these proofs at this time.

As to the graphical methods connected with beams and arches, the writer believes he has presented sufficient proofs for all the new parts of the method, merely referring to the basic equations which are found in all the works on these subjects.

Regarding the formula for determining the economical depth of plate-girders, the objection has been raised that it will consume too much time. You will notice that in the simpler forms, which cover more than nine-tenths of the applications, the determination of the depth automatically determines the area of the flanges; so that instead of being an extra operation it is simply a change in the method of attacking the old operation and—as the extraction of the square root to three places, which is all that is ever necessary, is a quicker operation than division—the designing of girders by this method is frequently quicker than by the older methods. The writer feels safe in saying that the average difference in time between this method and the old method will not be more than one minute at the outside.

The writer has a case in mind where a design of a proposed girder was submitted to him, and it was desired to strengthen it by about 15 per cent. The application of this method did not occupy more than a minute of time and resulted in not only strengthening the girder, but saving enough steel in three girders to be worth $65 in ordinary times; and $65 is certainly worth a minute of time, as engineers are paid now. In the more complicated forms, the time required is a little more but even the most complicated of these forms will not require more than five minutes in their application.

In the simple forms where no allowance is made for stiffeners, if the specification allows none of the web to be counted as flange, then the flange area is determined by multiplying the depth (from the formula) by the thickness of the web and dividing by two. If one-eighth of the web is counted as flange, then the division is by four, giving the area of one flange; one-eighth of the web being automatically deducted. By this last method results are reached more quickly, as an average, than by the old method.

To give an example of an application in a more complicated case—a girder in which the shear is 131 000 pounds and the moment is 19 822 000 inch-pounds. This will require stiffeners on the web, assuming the specifications by Chart 1 or by the usual railroad specifications. Assuming a ⅜-inch web and assuming that the stiffeners and fillers would add an additional 3/16 inch to this thickness (this last assumption is based on experience with girders under the above shear and moment):

The allowed fiber stress, f, = 16 000 pounds per square inch, and the thickness of web, t, = ⅜ inch, and t' = 3/16 inch; then $n = \frac{t + t'}{t} = 1.5$.

We will work this out on two assumptions.

First solution; none of the web being considered as flange:

$$y = \sqrt{\frac{2M}{fnt}} = \sqrt{\frac{2 \times 19\,822\,000}{16\,000 \times 1.5 \times \frac{3}{8}}} = \sqrt{4404} = 66.4$$
inches.

$\frac{ynt}{2} = 66.4 \times 9/16 \times \frac{1}{2} = 18.68$ square inches for each flange.

Web plate = 69 × ⅜.
Each flange consists of:

Two angles, 6 × 6 × 9/16	=	10.90 square inches,
and one plate, 15 × ⅝	=	8.28 square inches,
		19.18 square inches.

The web can be reduced to 66 × ⅜ inches.

Second solution; based on allowing one-eighth of the web to be counted with each flange:

$$y = \sqrt{\frac{2M}{ft\,(n - \frac{1}{4})}} = \sqrt{\frac{2 \times 19\ 822\,000}{16\,000 \times \frac{3}{8}\,(1.5 - 0.25)}} =$$
$\sqrt{5286} = 72.7$ inches. Web plate = 76 × ⅜.

$\frac{yt\,(n - \frac{1}{2})}{2} = 72.7 \times \frac{3}{8}\,(1.5 - 0.5)\,\frac{1}{2} = 13.65$ square inches = area of flange angles and cover plates if any are needed.

Each flange consists of two angles, 6 × 6 × ⅝ = 13.12 square inches. Increase web to 78 × ⅜ inches.

Of course experience has a great deal to do with the speed with which any method can be applied. It is not to be expected that the best results will follow the first attempts to apply this method, but the writer's experience shows that, in spite of the infinitesimal percentage that it may increase the time of calculation, it pays for itself a hundred fold. Of course we recognize the war-time conditions which have existed, but as we revert to a peace basis we must remember that where saving can be effected without extra expenditure of time, it pays.

THE APPLICATION OF ELECTRIC WELDING TO STEEL SHIPBUILDING

By H. A. Hornor*

INTRODUCTION

For many years electric welding methods have been applied to manufacturing and repair purposes in this country. The high temperature of the electric arc, though in the early days poorly controlled, gave undoubted promise to those who observed its properties. At that time scientific investigators were more interested in their endeavor to make a practical and economical lighting unit and the result was the development of the carbon arc lamp. As an offspring the carbon arc was suggested and is still employed as a process of welding metals. The step from the carbon arc to the metallic arc soon followed.

The use of the metallic arc in railway repair shops, as well as repairs to ship boilers, filling of the corroded parts of the hull, and other general repair work, has been quite general in this country for about 10 years. Until very recently electric welding was not considered applicable to the construction of new vessels.

The United States Shipping Board made arrangements with the British Admiralty for the visit of Captain James Caldwell, R. E., to this country for the sake of joining forces in the extension of this process to the vast shipbuilding program then undertaken by the two countries. Captain Caldwell made a survey of the extent of the employment of electric welding in this country. He has embodied his investigations and conclusions in a complete report which has just been published by the Emergency Fleet Corporation of the United States Shipping Board.

In this report is also embodied the status of the art in England, with an explanation of the underlying reasons for stimulating the application over there. The conditions in England were similar to those in the United States, and the only application of importance was the use of the electric arc for repair work.

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The method of spot welding was not as well known there as here. As the war proceeded, heavy demands were made upon the supply of oxygen. Much of the welding up to that time had been done with acetylene and other gas welding methods. The lack of oxygen was a serious matter and its use had to be curtailed. Investigations led to recommendations for the substitution of electric arc welding in order to conserve oxygen. Sufficient interest was aroused to cause Lloyd's Register of Shipping to make experiments, with a view to the establishment of special rules and requirements for the building of electrically welded steel vessels. These rules were recently approved at a special meeting of Lloyd's Committee at which were many of the most notable men connected with the British government, as well as representatives from Allied countries.

ELECTRIC WELDING

Aware of the many different kinds of welding processes, systems, and methods, the Welding Committee of the Emergency Fleet Corporation selected the two methods which seemed reasonably applicable to steel ship construction; namely, spot welding and arc welding. This must not be interpreted to mean that of the many other processes there are not some which may be equally reliable and equally beneficial to the shipbuilding industry but it has been considered that, as the application develops, other methods and processes will have ample opportunity for demonstration and trial.

The placing of two plates so that their edges lap, and the bringing of this jointure to a machine so equipped that the lapped plates will form a high resistance upon the short circuiting of the electric circuit under pressure is, briefly, the process of spot welding. This method was in general use on light materials in many industries in this country but it had not been extended to materials of the thickness of the mild steel plates forming the hull of a vessel. Experiments were undertaken a few months ago to determine whether there were fundamental limitations—either in the design of apparatus, or inherent in the electrical circuit—to the use of this process. A demonstration machine, capable of developing a current of approximately 100 000 amperes and having a pressure of approximately 20 tons per square inch, was

temporarily erected. The results with this machine conclusively proved that three thicknesses of one-inch plates could be satisfactorily and reliably joined by this method. A large stationary machine and two small portable machines specially designed for steel ship construction are now in service and a large portable spot-welding machine is in course of construction.

The process of electric arc welding, to employ the analogy used by one of the members of the Committee, is that of a small electric furnace. One side of a properly proportioned electric circuit is connected to the metals to be welded and the other side is connected to an iron rod manipulated manually. When the steel rod or electrode strikes the parent metal, an arc is sprung which instantaneously produces a temperature sufficient to fuse the parent metal and the electrode. In this manner the electrode material is deposited in the joint in the form, as nearly as can be ascertained to-day, of a steel casting. This fact brings the process into the scientific field of the metallurgy of steel.

The temperature of the electric arc is the highest known to man—3720 degrees C.

THE RIVETED STEEL SHIP

The first meeting of the Welding Committee established the fact that much saving in time and man power could be realized by the application of electric welding to a considerable part of the work of fitting up the riveted vessel. There were present at that meeting, representatives of the American Bureau of Shipping, and of Lloyd's Register of Domestic and Foreign Shipping—both of them Classification Societies. Inquiries were made as to what portions of the hull structure these Societies would at that time allow to be electrically welded. The reply was embodied in a document duly approved by the official representatives in this country. (See Appendix 1.)

This list consists mainly of parts which the Classification Societies do not strictly supervise, and which have been called non-essential parts of the structure. A liberal expansion of the list was expressly provided for, thereby justifying those who were assured of their ability to perform good work, in submitting any special designs to the Classification Societies for approval.

Some of the Eastern shipbuilders have, from their practical

experience, already extended the use of the electric arc and received approval from the Classification Societies for doing a very considerable portion of their work with this method. For instance, in addition to attachments to the hull, such as deck-rail stanchions; skylights; engine-room stairs and gratings; ladders, door-frames and casings; clips for attaching wood finishing; screened bulkheads; coal chutes; ventilator cowls, stacks and uptakes; cargo batten cleats; shaft alley escapes; etc., they are now permitted to weld electrically all bounding bars, masts, Otter gears, rivet heads, wrongly punched holes, hatch coamings, water and oil settling tanks, coffer-dams, etc. A single item—the bounding bars—represents a very large portion of the fitting up work in a standard riveted ship.

It frequently happens that a combination of a riveted joint and a calking weld is considered applicable. Such a design of joint is not reliable, due to the fact that in the working of the ship the riveted joint tends to loosen. In so doing it would throw strains or stresses upon the welded seam which it was not intended to withstand. If the welded portion of such a combination welded and riveted joint were made a strength weld there would be no reason for the rivets.

The difficulties encountered in maintaining the integrity of compartments in the ship, used exclusively for the carriage of fuel oil, made necessary some method to overcome the leaks occasioned by the aforesaid working of the riveted joint, as mechanical calking is not sufficient. In certain cases this was accomplished by placing a thin sheeting bent over the entire riveted lap-joint or butt-joint so that oil tightness would be maintained but the flexibility of the welded protection would permit the working of the riveted joint. This seems an absurdity except as an exigency repair measure, for in an original design entirely electrically welded joints would be better.

In contradistinction to the above discussion as regards the combination of a riveted joint and an arc-welded joint there is the successful union of the spot-welded joint with the arc weld. In this case the spot weld is a complete fusion of the parent metal, and the creeping action due to the working of the main structure does not exist and is impossible of occurrence. With such a com-

bination a calking arc weld along the edges provides assurance of perfect water-tightness eliminating entirely the question of corrosion, which is the most serious one connected with the riveted joint. This would permit of relatively wide spacing of spot welds and a reduction of overlap.

With all these factors in view, the ship designer is able to take a riveted ship design of the transverse type and primarily omit all staplings, cement, etc.; after notching all the inner channel flanges where cut by the deck to weld all decks to the shell. By using butt-weld seams and butts and vertical shell above the turn of the bilge there will be eliminated all laps, liners, joggling, straps, etc. This would still permit the riveting of plates to the frame. In like manner, there would be a similar saving if butt-weld seams were used for all deck plating in the same way as for the shell plating. By a combination of spot and arc welding of the brackets to either the beam or frame in the shop, these parts could then be arc welded to the other members in the ship. Similar treatment could be applied to the floor plates and the reverse frame clips. Straps, as now used in the construction of mast, booms, shaft and pipe tunnels, could be omitted and all cast-steel fittings as well as non-ferrous fittings safely attached by means of arc or spot welding. Such parts as wash plates may be tack welded to decks and bulkheads and all flanges omitted. In like manner, such appendages to the hull as bilge-keels, Otter gear, etc., would be better welded, as in this way the parent metal of the hull structure would not be impaired. It has been conclusively proved that the electric arc process does not injuriously affect the parent metal. As a matter of fact, in the structure of certain rolled-steel plates, metallographic indications as well as mechanical tests point to an improvement in the grain of the steel in the vicinity of the newly deposited electrode material. The effect shows that it does not go beyond one-sixteenth of an inch. In this connection, certain tests were made recently at the Norfolk Navy Yard to determine the welding of small fittings to special high-grade tensile steel. The results of these tests show that in all cases fractures occurred outside of the welded area, and that the effect of the heat was of no importance as far as the grain structure was concerned. "The zone forming the demarkation

between the welded and original metal shows a marked hardening but this disappears a very short distance away from this zone."

The practical electric welder has applied the process to hasten the construction of the riveted ship, with the result of large savings in time, money, and man power. In one of the Eastern assembling yards they are now doing 600 different jobs on each vessel. When this figure is multiplied by 28, 50, or 150 hulls it results in a very large item of work and this must be looked upon as merely a beginning of the application. In another Eastern assembling yard the total number of parts comprising the welding program for the standard riveted ships now under construction, amounts to 225 000 different pieces. The labor cost for riveting these pieces would amount to approximately $245 000, and for welding them about $99 000, making a total saving of $146 000. This saving represents somewhere in the vicinity of 70 per cent. In some cases the economy of the electric welding methods over riveting, rises to 90 or 95 per cent. There remains no question in the minds of those who are applying electric welding methods, that with proper facilities, well-trained operators, and the work properly prepared, time and money may be saved by this method.

THE ELECTRICALLY WELDED STEEL SHIP

The applying of electric methods to steel shipbuilding fell naturally into two parts when the matter was first taken up for discussion and the foregoing brief survey indicates to what extent this method has already made headway in the ship program. The second consideration is the possibility of constructing a ship without rivets and joining her framework together by means of electric methods. Investigations were made for the purpose of discovering whether electric welding had ever been largely used for this purpose. Recently attention was called to a 42-foot launch built in 1915 at Ashtabula Harbor, Ohio. This launch is self-propelled, has an 11-foot beam, is 6 feet 6 inches in depth, and is constructed of 8-pound steel plates, 11/64 (0.171) inch thick. The frames are ¼- by 1½-inch angles, and the keel is 1 by 5 inches. The frames and keel are riveted. The plates are butted, and welded with the metallic arc. Not only has this boat been in continuous service for three years but she has experienced what all vessels are liable to—severe shock in docking, and crush by

contact with heavier craft and by encountering heavy ice conditions. The following information regarding this boat is quoted from a letter from the owner and builder:

> "The launch has been in continuous operation since built having used same the year round and our experience with weld we consider far superior to riveting as we never had a leak in the weld and the rivets are sure to leak in a dented place."

A portion of the mid-ship body of a sea-going barge was welded in this country by means of the metallic electric arc at Newburg, New York, about a year ago, while the rest of the structure was riveted in the usual manner. This barge was designed for the purpose of carrying oil in bulk. She was fully loaded with a miscellaneous cargo and was towed from Newburg to Tampico, Mexico, a distance of approximately 2500 miles. Since her arrival in Tampico she has been used in regular service for transporting oil. She has been docked and examined and the electrically welded portions show no signs of deterioration.

Captain James Caldwell was instrumental in having built a cross-channel barge for carrying munitions to the British Expeditionary Forces. This vessel was 125 feet between perpendiculars and 16 feet in beam, with a displacement of 275 tons. The hull is rectangular in section amidships with only the bilge-plates curved. She was built up of 71 transverse frames and contains three bulkheads. The shell plating is one-fourth and five-sixteenths inch. The construction differs very little from the standard riveted type with lap-joints. Shell plates are arranged for clinker build and the plate edges are joggled. This latter arrangement was for the purpose of providing an easier method for accomplishing the electric welding in order to eliminate as much overhead welding as possible. Captain Caldwell reports that the keel was laid on February 15, 1918, and the vessel was launched on June 11, 1918, but the work was not continuous as it was necessary to call the men off for other more important labor. Therefore, considerable time was wasted and the summary of speed and cost suffers somewhat unduly on this account. He states that although for the first month the average speed per welder was only 4 feet an hour the speed of working was increased to as much as 14 feet an hour before the barge was

completed. "Taking all positions of work into consideration the average speed was 4 feet per hour at the commencement whilst toward the end of the work an average of 7 feet per hour was easily obtained."

This barge was erected in the same way as a regular riveted structure. Holes were spaced originally 10 inches apart, this being afterwards increased to 15 inches apart for service bolts for drawing the structure together. After the seams were welded, these service bolts were withdrawn and three methods were experimented with for plugging up the holes:

1. Punchings were hammered into the holes and welded over with an ordinary electrode.

2. The holes were entirely filled with deposited metal from a bare metal electrode.

3. Steel pins of correct length were inserted and welded up flush.

"None of these methods showed any superiority in time or ease of welding but the third was regarded as being superior in strength."

The total expenditure of money for the electric welding was $1500. The labor amounted to $310.

This barge has already encountered in service extremely severe conditions in the English Channel. She has been examined and, although some minor leaks were discovered after she was first loaded, there have been no signs of any conditions tending to condemn the process. Evidences of the success of this venture caused the immediate preparation of designs for a combination barge to be riveted and welded. In this design the floors will be riveted to frames, and beam-knees to frames and beams. The rest of the design provides for the welding of all other members of the structure. The underlying reason of the combination is due to the use in England of an extremely expensive type of covered metallic electrode. This is seen in the cost given above in which the labor, as previously mentioned, was $310, the electric current cost $300, and the electrodes $890. It will thus be seen that the electrodes cost more than the combined cost of electric current and labor. It was decided that by using rivets in certain [illegible] of this vessel they would be able to offset the extreme

cost of the electrode material. In the United States the practice has been to use the bare electrode which is very much less expensive than the covered.

Without entering into a lengthy discussion of the comparative advantages of these two proceesses it will suffice to state that a coating of some kind seems to improve the ductility of the finished weld. It is not to be expected in this country that the accepted electrode material will be as expensive as that now preferred in England. Undoubtedly, for the strength members of a vessel it will be necessary to use the best material and the best workmanship that can be procured. On the many "non-essential" parts of the structure, American engineers are convinced that it will not be necessary to expend as much money on electrodes.

In dealing with riveted connections it was first observed by Mr. E. H. Ewertz that it was necessary to use angle bars. The welded joint can be made by placing two plates or bars at right angles and completing the jointure by welding the seams. Mr. Ewertz made this the basis of a patented construction for barges. His patent includes the flanging of one edge of the plate and in this way he combines the obvious advantages of both the transversely framed and longitudinally framed constructions. By cutting notches, either mechanically or by means of the oxy-acetylene torch, in the straight edge of the plate he is able to insert a bolt and thus assemble the ship in the usual manner. These notches can be electrically welded after the main structure has been welded. This design requires the flanging of the plates at the mill as the present materials do not lend themselves to cold flanging. The design merits careful consideration and it can be adapted to larger vessels.

Early in the deliberations of the Welding Committee it was proposed that a special design of an electrically welded ship be prepared. A sub-committee was appointed for this purpose in order to submit designs for the approval of the Emergency Fleet Corporation. This work was done under pressure as the limitation of time was then considered as of the utmost importance, and it is to the credit of those who handled this work, that complete plans and an outline description were presented within three

weeks. These designs not only included the ship herself but also drawings for a proposed yard in which to build her. At that time the general desire was for the large-sized cargo carrier and the plans were laid out for a 9200-ton standard vessel. Those concerned with the design desired to combine the opinions of the two existing schools of thought—namely, conservatives and radicals.

It was decided that the frames should be riveted to the beams and to the floors. Transverse framed construction was considered to be the best basis of design for electric welding. Only standard shapes were to be employed and a little wider plate was to be used in order to assist in what was probably the most radical part of the design—namely, vertical plating instead of the usual horizontal plating. The reason for adopting this idea has to be considered not only in connection with the design of the vessel but in conjunction with the suggested method of building. The proposed system of building was on the basis of six-foot welded sections. These sections were to be fabricated in six pieces in the shop, transported to the ways, and put together by means of what was called a "header." This structure was arranged on tracks to span the ways and to move up the ways from stern to stem. The structure was designed to carry two electrically driven 46-foot cranes, and was divided into six decks. These decks would contain all the necessary welding apparatus and the platform or staging upon which the operators would work. The calculations were based on the completing of one six-foot section each eight-hour day. It was estimated that with a single shift the vessel could be completed, all equipped, and ready for sea in 75 working days. This estimate included a 10 per cent. margin.

The dimensions determined upon were: 400 feet between perpendiculars, breadth 54 feet, depth 32 feet 10 inches. There were two decks, poop, bridge and forecastle, and the usual top hamper with deck gear and cargo handling appliances.

The plates were butted at every alternate frame. The frames were spaced 36 inches. Every alternate channel frame, therefore, made a backing strip for the butted plates, producing a strap joint. This type of joint gives three seams of welding—

one between the butted plates, one at the heel and the other at the toe of the channel frame. This type of welded joint is considered to be a joint of 100 per cent. efficiency or over. Great care was exercised in the designing of these joints and in their use throughout the structure. Careful calculations were made and increased scantlings employed wherever considered requisite as an additional safety factor for the use of electric welding.

The modulus of resistance of a mid-ship section of the vessel, taken in the way of a cargo hatch and through a butt line of the transverse plating, showed it to be 10 per cent. in excess of the riveted vessel. This is most conservative in view of international tests showing riveted joints of the greatest strength only 75 per cent. efficient in comparison with the weakest electrically lap-welded joint of about the same efficiency. The saving in steel due to the redistribution of materials—i. e., overlap in plating and weight of rivets—gives the welded ship design 500 tons more cargo capacity (or deadweight) on the same draft, than is obtainable with the identical ship as at present being built.

This ship design in conjunction with the yard designed for her building indicates the way that is open for the development of rapid and economical steel shipbuilding construction. The basis of the design of ship and yard was to provide a maximum of work in enclosed and comfortable shops where special craftsmanship, which is the art of electric welding, could be carried on by the best experts obtainable. This means a very different type of labor from any that has yet been introduced into shipbuilding. Undoubtedly the work could be so arranged and the conditions so adjusted that the work would permit of the use of women for all but the heavy handling of materials.

This design reduced to a minimum the number of crane lifts at the assembling yard—the principal difficulty now encountered in the rapid erection of steel ships. These designs provided the minimum amount of welding in the field and assured a quick method not only of manufacturing a ship but of advancing its progress by equal and decided increments. It must be understood that this scheme carried with it a practical idealism for the assembling of all the accessories which are necessary for such a complex combination. It can readily be seen that there must first

be a careful organization to lay out the proper delivery of the various parts of the ship so that the "header" may move from stern to stem without interruption.

The proposed designs both in England and in the United States have been greatly aided by the results of important tests of electric welding recently announced by Lloyd's Register of Shipping. In England Messrs. Cammell, Laird & Company are now engaged in construction of an electrically welded coastwise ship. It is understood that the vessel will have a carrying capacity of 1000 tons. She will be self-propelled by a Cammell, Laird-Fullagar oil engine—a new type of ship propulsion. The interesting point here will be the trial of a large-sized, self-propelled, electrically welded, steel ship.

About a month ago Mr. J. W. Isherwood brought over to this country a set of plans devised by him in collaboration with Mr. Abell of Lloyd's Register for an electrically welded ship. Mr. Isherwood is the originator of a patented system of longitudinally framed vessels. His design of welded ship follows the same principle and has the added advantage of largely reducing overhead welding. In fact there is no overhead welding necessary for any of the strength joints. He has maintained the use of holes spaced approximately 12 inches apart, which permits the customary method of assembling the ship both for the purpose of drawing the members together and for fairing up the various sections. Mr. Isherwood is of the opinion that it would probably be as economical to join the structure through the punched holes with rivets as to fill up the holes afterwards by means of the electric arc. In certain parts of the design, where it has been necessary to space the holes nearer together for the purpose of assembly, he believes that the few additional holes required for the riveted joint may as well be added and rivets used. His general point of view is that of conservatism in erecting and assembling the vessel but he is optimistic as to the saving and the perfection to be secured by the use of the electric arc method. It is understood that one vessel of this design is now being projected in England and that several more of larger size are soon to be laid down.

All future designs will undoubtedly follow the suggestions of Lloyd's Register of Shipping. A copy of the Rules is appended to this paper. (See Appendix 2.) The basis upon which Lloyd's adopted these tentative regulations was a series of tests made with large-sized specimens of welded joints. The results were as follows: Butt-welds showed a tensile strength carrying from 90 to 95 per cent. of the tensile strength of the unwelded plate. Riveted lap-joints, with plates of about one-half inch in thickness, averaged about 65 to 70 per cent. of the strength of the unperforated plate. The electrically welded lap-joint, with full fillet welds on both edges, showed an ultimate strength in tension bearing from 70 to 80 per cent. of that of the unwelded material. With a full fillet and a single run of weld on the other edge the results were very little inferior.

Another interesting observation from these experimental results is that of dynamic elasticity. Regarding this Lloyd's Register states as follows:

"In a structure, such as a ship, which is exposed to variations and reversals of stresses, it is extremely important to know whether the material to be used is likely to break down rapidly under such alternations and ranges of stress as are likely to be experienced. The modified Wohler tests employed in the experiment certainly indicate, if considered solely by themselves, that whereas for a given number of alternations mild steel would withstand a range of stress, of say, plus or minus 10½ tons, the welds might be expected to fail at about plus or minus 6½ tons, a figure which is more nearly experienced in ordinary ship construction."

CONCLUSION

Appendix 3 gives the results of a series of tests made by the United States Bureau of Standards on a number of samples of arc-welded one-half-inch steel plates of the composition at present used in the shipbuilding industry in the United States. The results of these tests conclusively show that this method is satisfactory for the building of large-sized cargo vessels. These tests will be followed by similar ones made with three-quarter-inch and one-inch plates. The average tensile strength as shown by these results is very nearly the same as the tensile strength of the parent metal and many of the samples fractured outside of the weld.

The following facts, though briefly stated, settle any doubt as to the reliability of the electric welding methods for the application proposed:

1. Spot welding of plates of the sizes necessary can be made in a satisfactory manner.

2. The skilled arc-welding operator can produce reliable welds with any system for the control of the electric circuit.

3. The quality of the weld depends more on the skill of the operator and the choice of the electrode and current than upon any other factor.

4. Successful welds are the result of the holding of a short arc.

5. Apparatus which maintains a short arc, whether on alternating or direct-current systems, though requiring more skill on the part of the operator, is in general the safest method of providing reliable welds.

6. Satisfactory welds can be made with either alternating or direct current.

7. Bare and covered electrodes produce welds of approximately equal tensile strength, but the covered or coated electrode weld is usually more ductile than the bare electrode weld.

8. The relative efficiencies of the three usual joints made with the electric arc are: Lap-joints 75 to 80 per cent.; butt-joints 90 to 95 per cent.; strap-joints 100 per cent. or over.

9. The riveted joint with water-tight spacing shows an efficiency of approximately 75 per cent., about equal to the electrically welded lap-joint.

These facts, with the results of work now performed, are conclusive evidence as to reliable performance of the electric welding method of joining the materials for the construction of steel ships.

APPENDIX 1

The Classification Societies have so far considered and approved of the application of electric welding to the following parts of vessels:

Deck rail stanchions to plating.

Clips for detachable rail stanchions.

Continuous railing rods (joints).

Attaching deck collar (L-rings) around ventilators.

Attaching deck collar (L-rings) around smoke stack.

Attaching cape rings, smoke stacks, pipes, etc.

Attaching galley fixtures to plating.

Attaching bath and other fixtures in officers' quarters.

Attaching cowl supporting rings to ventilators.

Bulwark rail top splicing and end fittings.

Skylights over galley.

Engine-room stairs and gratings.

Boiler-room stairs and gratings.

Attaching engine- and boiler-room stairs and gratings to plating grab rods on casing.

All stairs and ladders, including rail attachments.

Door-frames to casing; hinges, catches, holds, coach hooks, etc.

Clips for attaching interior wood finish to casing.

Entire screen bulkhead; also coal chutes.

Butt of water-tight and oil-tight boundary bars on bulkheads or floors in double bottom. Ventilator cowls, stacks and uptakes.

Bulkheads (not structural parts of the ship, partition bulkheads in accommodation).

Framing and supports for engine- and boiler-room flooring or gratings.

Cargo batten cleats.

Tanks (not structural parts).

Shaft alley escapes.

Steel skylights over accommodation spaces.

Engine-room skylights.

Grab rods on exterior and interior of deck houses.

Deck houses not covering unprotected openings through weather decks.

Reinforcing and protecting angles around manholes.

Joints of water-tight angle collars at frames in way of water-tight flats.

Other parts of a vessel in which electric welding is proposed must be submitted for consideration. March 25, 1918.

For Lloyd's Register of Shipping, J. French.
For American Bureau of Shipping, G. G. Sharp.

APPENDIX 2

Tentative regulations for the application of electric arc welding to ship construction.

A. System of Welding and Workmanship.

1. The system of welding proposed to be used must be approved and must comply with the regulations and tests laid down by the Committee.

2. The process of manufacture of the electrodes must be such as to ensure reliability and uniformity in the finished article.

3. Specimens of the finished electrodes, together with specifications of the nature of the electrodes, must be supplied to the Committee for purposes of record.

4. The Committee's officers shall have access to the works where the electrodes are manufactured and will investigate from time to time, as may be necessary, the process of manufacture to ensure that the electrodes are identical with the approved specimens.

5. Alterations from the process approved for the manufacture of electrodes shall not be made without the consent of the Committee.

6. The regulations for the voltage and amperage to be used with each size of electrode, and for the size of electrode to be employed with the different thicknesses of material to be joined, are to be approved by the Committee.

7. The Committee must be satisfied that the operators engaged are specially trained, and are experienced and efficient in the use of the welding system proposed to be employed.

8. Efficient supervisors of proved ability must be provided, and the proportion of supervisors to welders must be submitted for approval.

B. Details of Construction.

9. The details of construction of the vessel and of the welds are to be submitted for approval.

10. Before welding, the surfaces to be joined must be fitted closely to each other and the methods to be adopted for this purpose are to be approved.

11. All butt and edge connections are to be lapped or strapped.

12. With lapped connections the breadths of overlaps of butts and seams and the profiles of the welds are to be in accordance with the following table:

Thickness of plate in inches	Width of overlap seam and butt in inches	Throat thickness in inches
0.40 and under	2¼	0.28
0.60	2½	0.38
0.80	2¾	0.48
1.00	3	0.50

Intermediate values may be obtained by direct interpolation, and for thicknesses below 0.40, the throat thickness is to be about 70 per cent. of the thickness of the plate.

13. A "full weld" extends from the edge of a plate for a distance equal to the thickness of plate to be attached, and the minimum measurement from the inner edge of plate to the surface of weld is the throat thickness given in the table above.

14. A "light closing weld" is a single run of light welding continuously along the edge of the plate. Such a weld may, however, be interrupted where it crosses the connection of another member of the structure.

15. An "intermittent or tack weld" has short lengths of weld which are spaced three times the length of the weld from center to center of each short length of weld. Such tack welding may vary in amount of weld between a "full weld" and a "light closing weld."

16. The general character of welds is to be in accordance with the following table:

	Inside edge	Outside edge
a. Butts of shell, deck and inner bottom plating. b. Butts of longitudinal girders and hatch coamings.	F	F
c. Edges of shell, deck and inner bottom plating. d. Butts and edges of bulkhead plating.	L	F
	Toe	Heel
e. Frames to shell, reverse frames to frames and floors. f. Beams of decks. g. Longitudinal continuous angles. h. Side girders, bars to shell, intercostal plates, floors and inner bottom. i. Bulkhead stiffeners.	T	L

(F = full weld, L = light weld, T = tack weld.)

17. All bars required to be water tight are to have continuous welding on both flanges with tack welding at heel of bar.

18. The welded connections of beam, frame, and other brackets, are to be submitted for special consideration.

19. The Committee may require, when considered necessary, additional attachment beyond that specified above, and the welding of all other parts is to be to their approval.

U S Shipping Board
Emergency Fleet Corporation
Electric Welding Committee
Plates Tested by Division VII-1 and VIII
Bureau of Standards - Washington, DC

Test of Plate Material

1	2	3	4	5	6	7
Tensile Test			Torsion Test		Vibration Test	Bending Test
Yield Point in Lbs per Sq In	Elongation in 2 in. per %	Ultimate Strength Lbs per Sq In	Angular Degrees of Plate	Torque of Plate in Radius Lbs	No of Oscillat. of Plate	Angular Degrees of Plate
38400	.115	64700	574	6048		O K for 180°

Weld

8 Serial No of Current Test	11 Operator	Position: Flat	Horiz	Vertical	Overhead	16 Type	17 Rate in Ft Hours	20 Electrode Type	21 Diameter in Inches	22 Manufact'r of Electrode	27 Power: Current in Amperes AC	28 DC	29 Voltage: Open Circuit	30 Arc	33 Remarks on Weld
	R B	·					333		15625	Roebling		170-180	67-68	18-22	Each weld made … Plates cleaned by sand blast
.	.	✓					286		.	-		150-160	67-68	18-22	Each weld made in 1 run Plates cleaned by sand blast
3	GOW	·					235	Am Basic Annealed	.	Am Steel & Wire		110-120	63	14-17	
4	.	✓					364	"	1875	-		140	63	15	
5							444		15625	.		140-150	75	18-20	Each weld made in 2 runs
6							400		.	.		140-150	75	18-20	Each weld made in 2 runs
7	TCT			✓			421		.	Roebling		140	60	15	
8	.			✓			400		.	"		140	60	15	
9A	Hines						400		.		140 60 Cycle		220	20	
9	SEP	✓					333	Tenson	.	Western Wire Co		150	75	15-18	
10	ACC	✓					320	.	.	-		150	75	15-18	
11	HWS	✓					181		.	Roebling		140	376	18-22	Each plate welded with 2 runs on 1st side 1 run on 2nd side
12	.	✓					250		.	-		140	375	18-22	Each plate welded with 2 runs on 1st side 1 run on 2nd side
13	PK	✓					332	Diamond Welding Wire	.	S & W Co		150-160	63	24-28	
14	-	✓					285	.	.	"		145-155	63	20-22	
15	DOT	✓					154	Coated	134	Q A		90	110	35	Weld made in 2 runs
16	-	✓					181	.	-	-		90	110	35	.
17	JMAI	✓					143	.	.	.	90		102	35	.
18		✓					181	.	.	.	90		102	35	.
19	TJ						190	.	125	E AC & W Co	150 60 Cycle		115	20-25	Weld made in 2 runs Current used 126 K W Hrs
20	.						148	Bare	125	.	175 60 Cycle		125	15	Weld made in 2 runs Current used 16 K W Hrs
21	T T	✓					16		134	Q A	105 25 Cycle		100	40	Input 5 0 K W - 12 7 K W Hrs
22	J J	✓					97	Coated	125	E A C & W Co	160 25 Cycle		100	20	Operator's first experience with 25 cycle and lost considerable
23	ELC	✓					210		.	Roebling		80-90	63	18-20	time cleaning out slag and changing electrode. Weld made
24	JWF	✓					190		-	-		80-85	63	18-20	in 2 runs the arc was considerably spattering when making input 35 K W
25	.	✓					320		15625	.	136 60 Cycle		115	18-20	
26	-	✓					266		.125	.	94 60 Cycle		230	17	
27	HKB	✓					285		.	.		110	78	20-22	
28		✓					266		-			110	78	20-22	
31	Gaston	✓					384		166	Roebling		150		20	Welded 1 run each side
32	-						312		-	.		150		20	Welded 2 runs each side

Tests

8 Serial No	38 Tensile Test: Yield Point Lbs per Sq In	39 Elon in 2 in Per cent	40 Ultimate Lbs per Sq In	41	42 Torsion Test: Weld Angular Degrees	43 Ratio in Per cent	44 Weld Torque in In Lbs	45 Ratio in Per cent	46 Vibration Test: Calculated Fibre Stress Lbs per Sq In	47 No of Oscillations of Weld	48 Ratio in Per cent	49 Bending Test: Weld Angular Degrees	50 Ratio in Per cent	51 Testers Serial No	52 Remarks on Test
1	35400	8.0	59800		443	77.2	5338	88.3				42			
2	39000	11.5	62600		464	80.8	5512	91.1				44			
3	36300	13.0	65470	•											
4	34300	5.0	50600		322	56.1	4728	78.1				34			
5	35300	7.0	52280	•											
6	35650	8.5	61100		306	53.3	4820	76.4				55			
15	42440	8.0	57340	•											
16	40280	4.0 52.5	39400 58740	•	261	45.5	4080	57.5				30			
17	38400	7.00	50250		[illegible]	76.8	4788	[illegible]				35			
18	40480	11.0	61760	•											
19	37500 37000	8.0 8.0	54000 65400	•	[illegible]	[illegible]	3900	64.5				[illegible]			
20	37200 42200	12.0 11.0	62800 66480	•	320	55.8	4332	71.6				38			
21	42400	9.0	62700		425	74.0	5381	88.9				34			
22	30700 39000	5.0 9.0	42200 66400	•								42			
23	37100	6.0	50000		274	47.7	4596	76.0				78			
24	46440 37850	5.0 4.0	57060 51900	•	238 249	41.5 43.4	4340 4176	71.7 69.1							
25	41800	6.0	45400		191 148	33.3 25.8	4296 3541	71.0 58.5	25000	36981		15			
26	44700 35300	4.0 3.5	41800 43700	•	245	41.2	4128	68.2				26			
27		2.0	34200		101 108	17.6 18.8	3472 3177	57.4 52.5	25000	123170		15			
28	31100 30600	7.0 8.0	48200 45800	•	465	81.1	4668	77.2				42			
31	38500	8.0	60900		518 450	90.3 78.4	6288 5856	104.0 96.8							Fractured 3? …
32					490	85.4	5568	92.1	25000	5376…		71			

Note. Columns 9, 10, 18, 19, 23, 24, 25, 26, 31, 32, 34, 35, 36, 37 omitted due to lack of data

Note Yield Point from Beam Drop
Weld not machined marked •

APPENDIX 3

Tests of ½-inch structural steel ship plates. Sample weld.

1

uous

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brack

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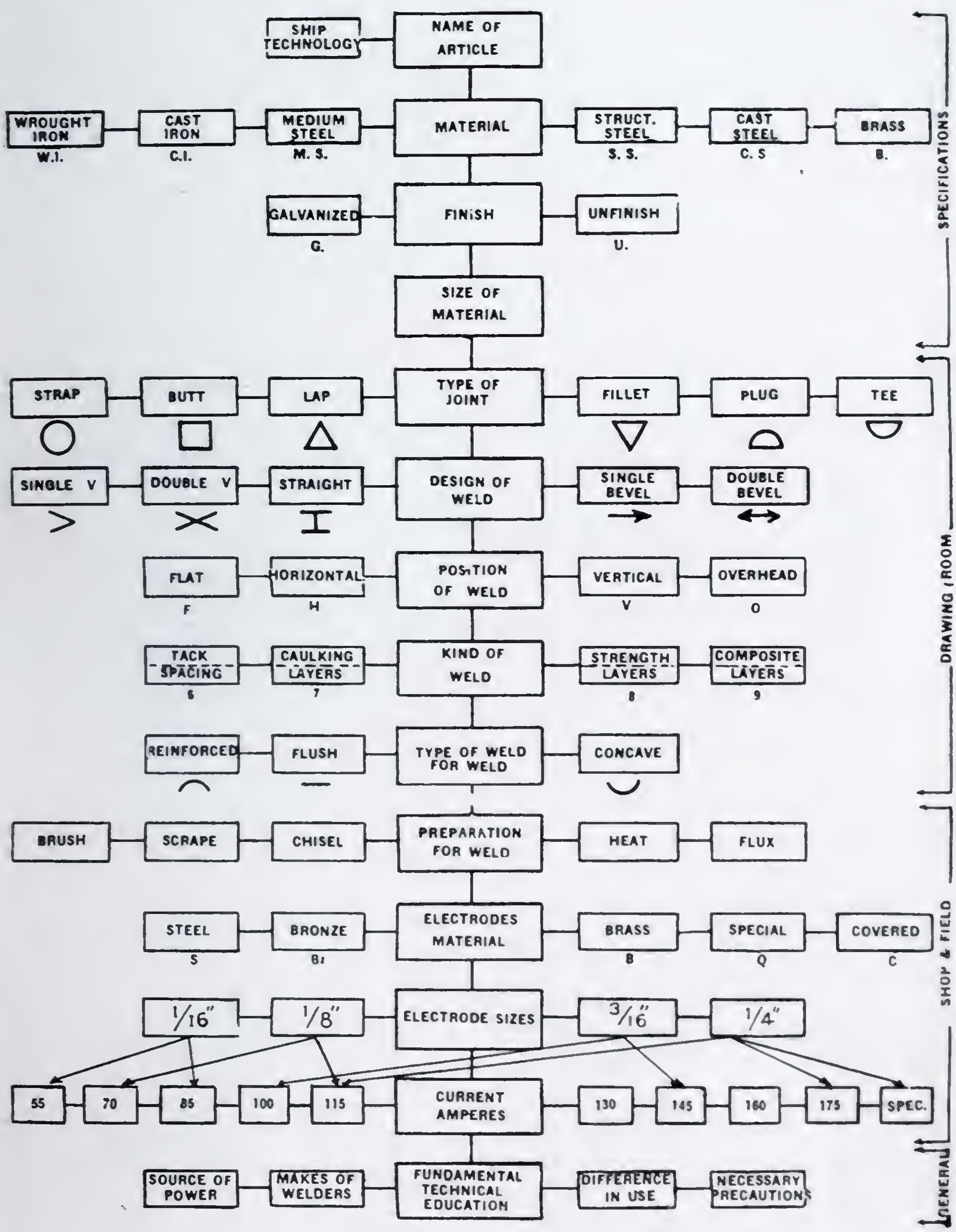
sary,

weldi

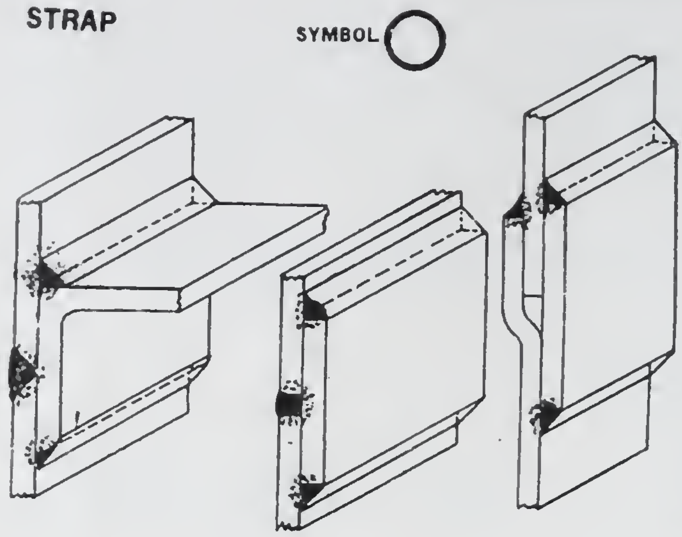
APPENDIX 4

Set of standard symbols and nomenclature prepared by the Electric Welding Branch, Education and Training Section. United States Shipping Board, Emergency Fleet Corporation.

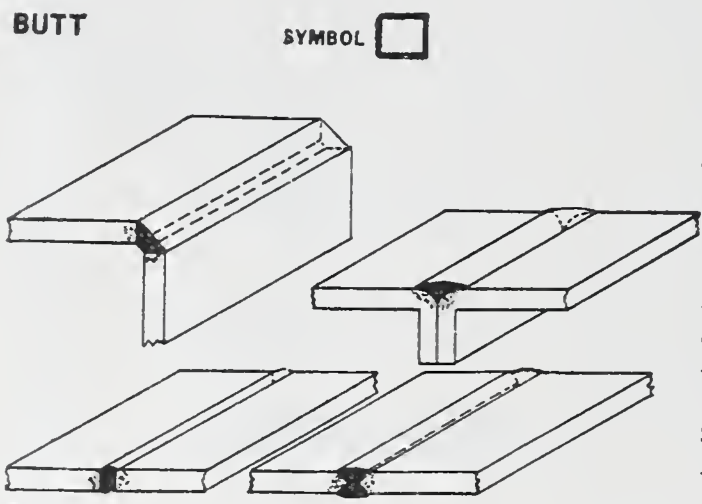
INSTRUCTION CHART WITH STANDARD SYMBOLS



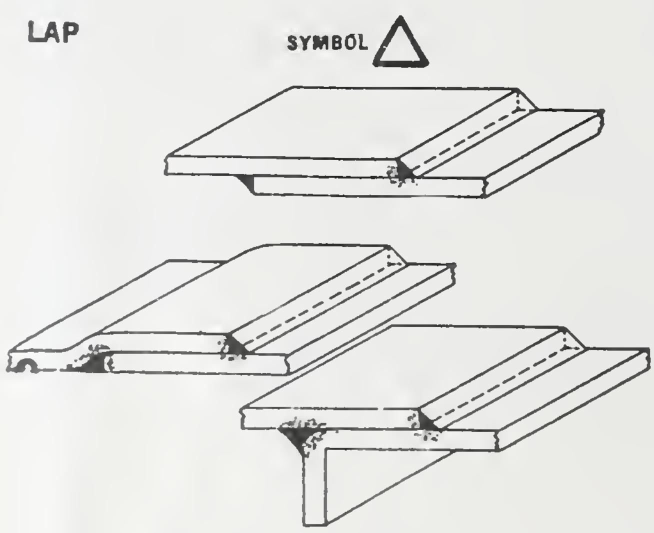
TYPE OF JOINT



Strap weld is one in which the seam of two adjoining plates or surfaces is reinforced by any form or shape to add strength and stability to the joint or plate. In this form of weld the seam can be welded only from the side of the work opposite the reinforcement, and the reinforcement of whatever shape must be welded from the side of the work to which the reinforcement is applied.

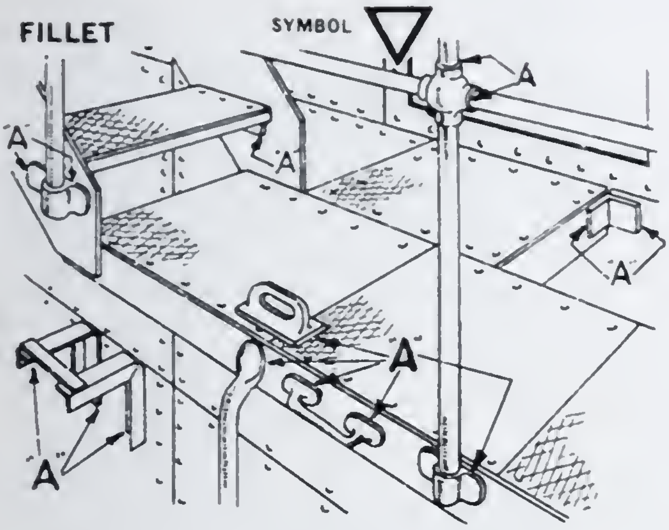


Butt weld is one in which two plates or surfaces are brought together edge to edge and welded along the seam thus formed. The two plates, when so welded, form a perfectly flat plane in themselves; excluding the possible projective caused by other individual objects as frames, straps, stiffeners, etc., or the building up of the weld proper.

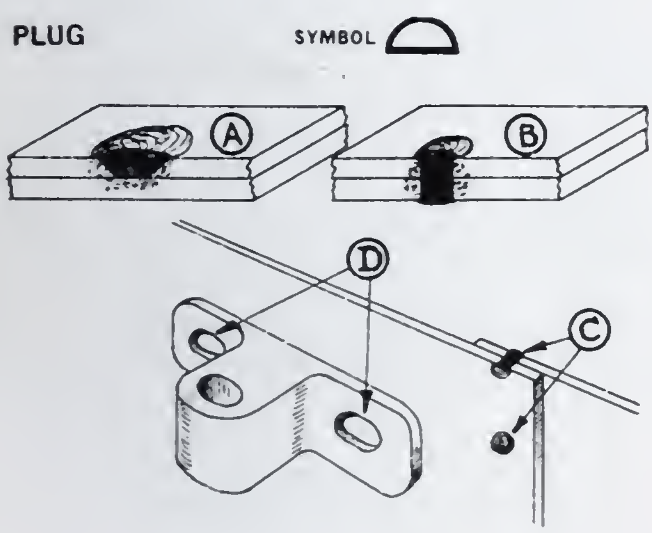


Lap weld is one in which the edges of two plates are set one above the other and the welding material so applied as to bind the edge of one plate to the face of the other plate. In this form of weld the seam or lap forms a raised surface along its entire extent.

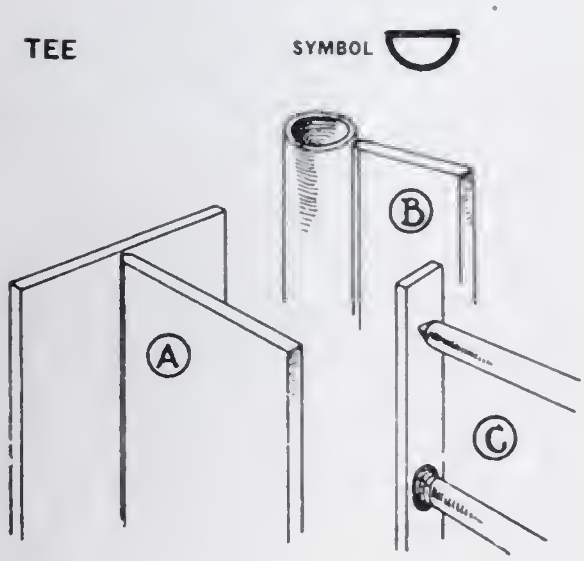
TYPE OF JOINT



Fillet weld is one in which some fixture or member is welded to the face of a plate, by welding along the vertical edge of the fixture or member (see welds shown and marked A on illustration at left). The welding material is applied in the corner thus formed and finished at an angle of forty-five degrees to the plate.

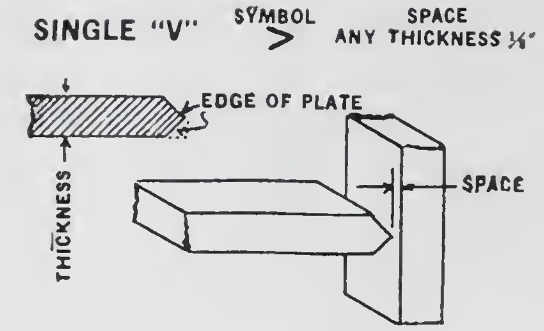


Plug weld is one used to connect the metals by welding through a hole in either one plate (A) or both plates (B). Also used for filling through a bolt hole as at C, or for added strength when fastening fixtures to the face of a plate by drilling a countersunk hole through the fixture D, and applying the welding material through this hole, as at D, thereby fastening the fixture to the plate at this point.

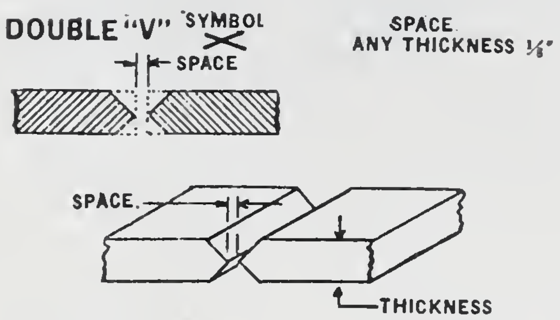


Tee weld is one in which one plate is welded vertically to another as in the case of the edge of a transverse bulkhead (A) being welded against the shell-plating or deck. This is a weld which in all cases requires *exceptional* care and can be used only where it is possible to work from both sides of the vertical plate. Also used for welding a rod in a vertical position to a flat surface, as the rung of a ladder (C), or a plate welded in the case of water-closet stalls.
vertically to a pipe stanchion (B), as

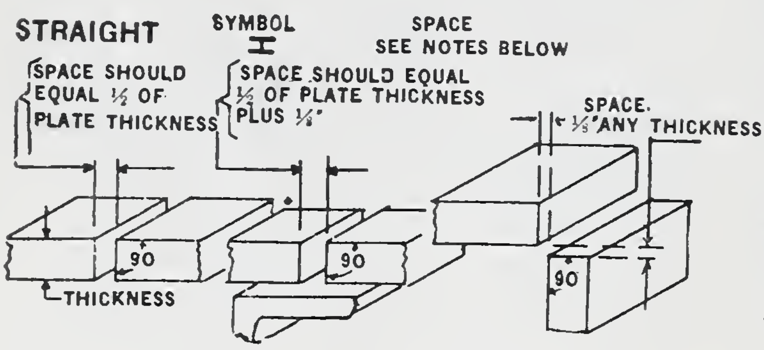
DESIGN OF WELD



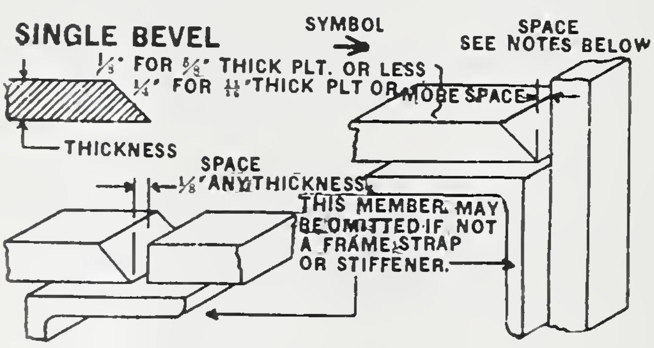
Single "V" is a term applied to the "edge finish" of a plate, when this edge is beveled from *both* sides to an angle, the degrees of which are left to the designer. To be used when the "V" side of the plate is to be a maximum "strength" weld, with the plate set vertically to the face of an adjoining member; and only when the electrode can be applied from both sides of the work.



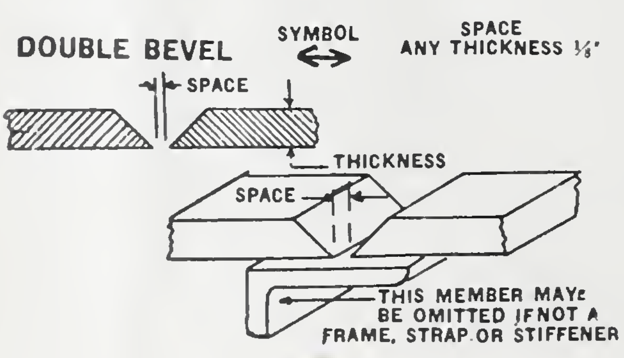
Double "V" is a term applied to the "edge finish" of two adjoining plates when the adjoining edges of both plates are beveled from *both* sides to an angle, the degrees of which are left to the designer. To be used when the two plates are to be "butted" together along these two sides for a maximum "strength" weld. To be used only when welding can be performed from both sides of the plate.



Straight is a term applied to the "edge finish" of a plate, when this edge is left in its crude or sheared state. To be used only where maximum strength is *not* essential, or unless used in connection with strap, stiffener or frame, or where it is impossible to finish the edge otherwise. Also to be used for a "strength" weld, when edges of two plates are set perpendicularly to each other—as the edge of a box.

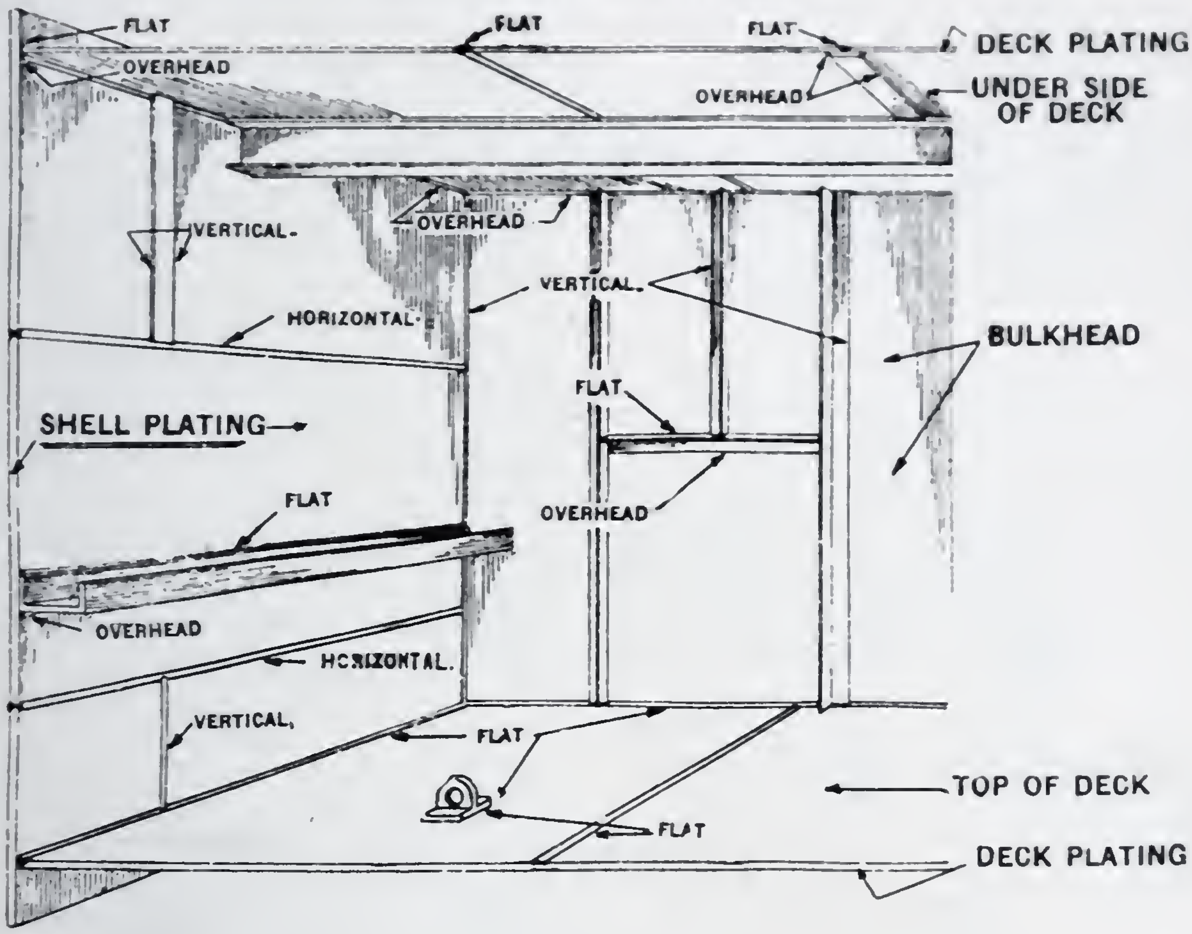


Single bevel is a term applied to the edge finish of a plate, when this edge is beveled from *one* side only to an angle, the degrees of which are left to the designer. To be used for "strength" welding, when the electrode can be applied from *one* side of the plate only, or where it is impossible to finish the adjoining welding surface.



Double bevel is a term applied to the edge finish of two adjoining plates when the adjoining edges of both plates are beveled from *one* side only to an angle, the degrees of which are left to the designer. To be used where maximum strength is required, and where electrode can be applied from *one* side of the work only.

POSITION OF WELD



Flat position is determined when the welding material is applied to a surface on the same plane as the deck, allowing the electrode to be held in an upright or vertical position. The welding surface may be entirely on a plane with the deck, or one side may be perpendicular to the deck and welded to an adjoining member that is on a plane with the deck.

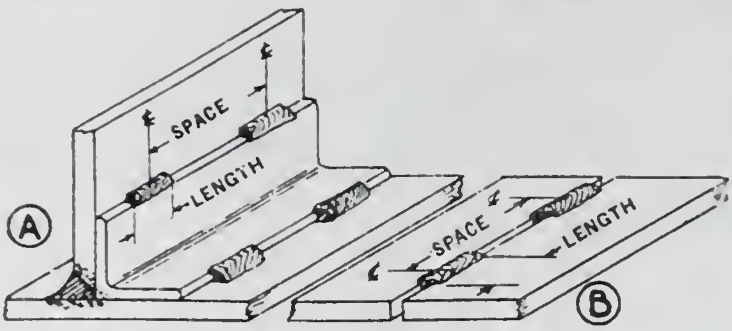
Horizontal position is determined when the welding material is applied to a seam or opening, the plane of which is perpendicular to the deck, and the line of weld is parallel with the deck, allowing the electrode to be held in an inboard or outboard position.

Vertical position is determined when the welding material is applied to a surface or seam whose line extends in a direction from one deck to the deck above, regardless of whether the adjoining members are in a single plane or at an angle to each other. In this position of weld, the electrode would also be held in a position partially parallel to the plane of the work.

Overhead position is determined when the welding material is applied from the under side of any member whose plane is parallel to the work and necessitates the electrode being held in a downright or inverted position.

KIND OF WELD

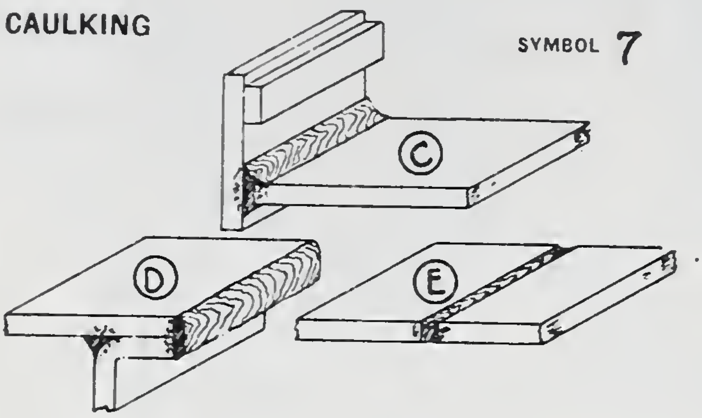
TACK SYMBOL 6



A *Tack* weld is applying the welding material in small sections to hold two edges together, and should always be specified by giving the *space* from center to center of weld and the *length* of the weld itself. No particular "Design of weld" is necessary of consideration.

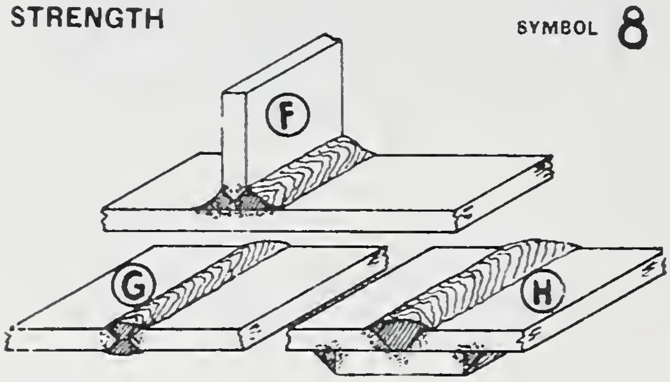
A *tack* is also used for temporarily holding in place material that is to be solidly welded, until the proper alignment and position are obtained; and, in this case, neither the *length, space*, or *design* of weld is to be specified.

CAULKING SYMBOL 7



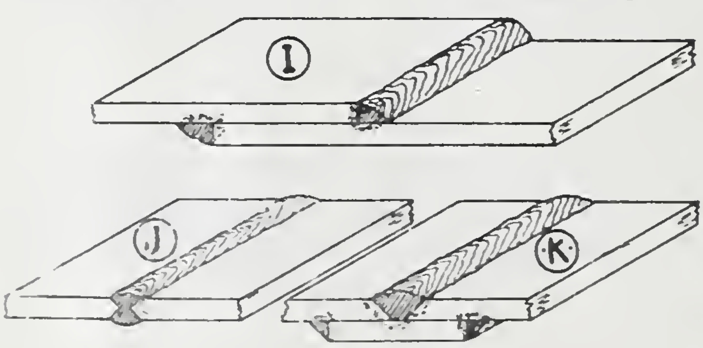
A *Calking* weld is one in which the density of the crystalline metal, used to close up the seam or opening, is such that no possible leakage is visible under a water-, oil-, or air-pressure of 25 pounds per square inch. The ultimate strength of a calking weld is not of material importance, neither is the "Design of weld" of this kind necessary of consideration. The operator must be the judge of the number of layers needed for a tight weld, although the designer should specify a minimum number of layers.

STRENGTH SYMBOL 8



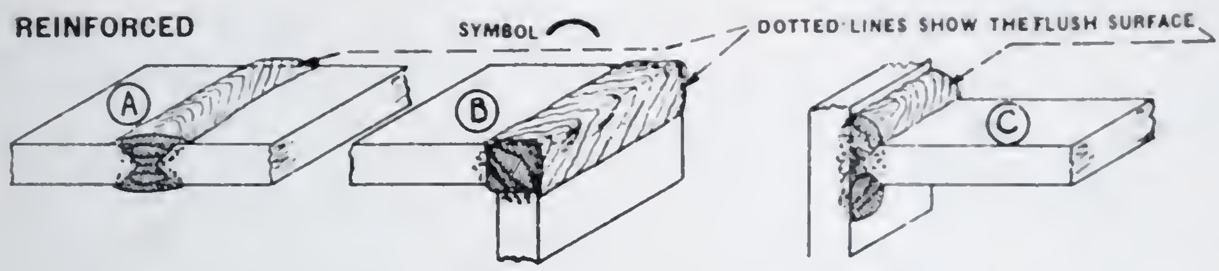
A *Strength* weld is one in which the sectional area of the welding material must be so considered that its tensile strength and elongation per square inch must be equal to at least 80 per cent. of the ultimate strength per square inch of the surrounding material (to be determined and specified by the designer). The welding material can be applied in any number of layers beyond a minimum specified by the designer. The density of the crystalline metals is *not* of vital importance. In this form of weld, the "Design of weld" must be specified by the designer and followed by the operator.

COMPOSITE SYMBOL 9

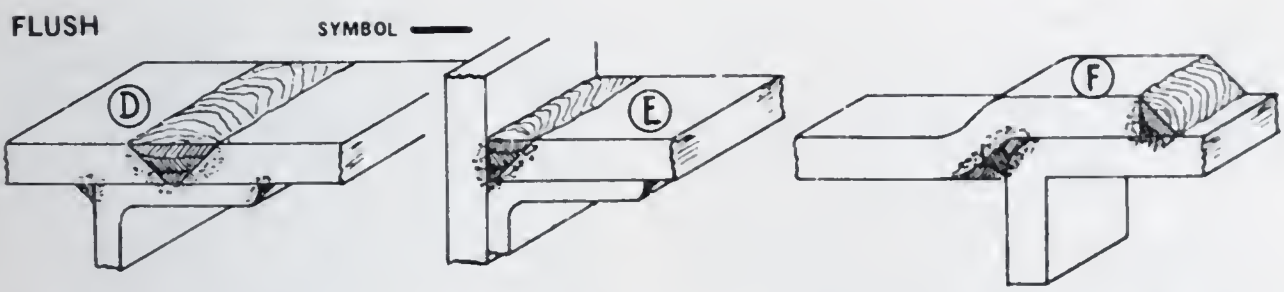


A *Composite* weld is one in which both the strength and density are of the most vital importance. The *strength* must be at least as specified for a "strength" weld, and the density must meet the requirements of a "calking" weld both as above defined. The minimum number of layers of welding material must always be specified by the designer, but the welder must be in a position to know if this number must be increased according to the welder's working conditions.

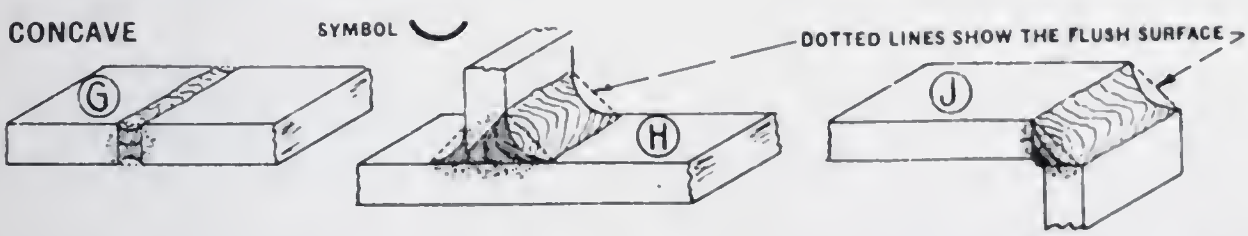
TYPE OF WELD



Reinforced is a term applied to a weld when the top layer of the welding material is built up above the plane of the surrounding material as at A or B above, or when used for a corner as in C. The top of the final layer should project above a plane of 45 degrees to the adjoining material. This 45-degree line is shown "dotted" in C above. This type is chiefly used in a "strength" or "composite" kind of weld for the purpose of obtaining the maximum strength efficiency, and should be specified by the designer, together with a minimum number of layers of welding material.

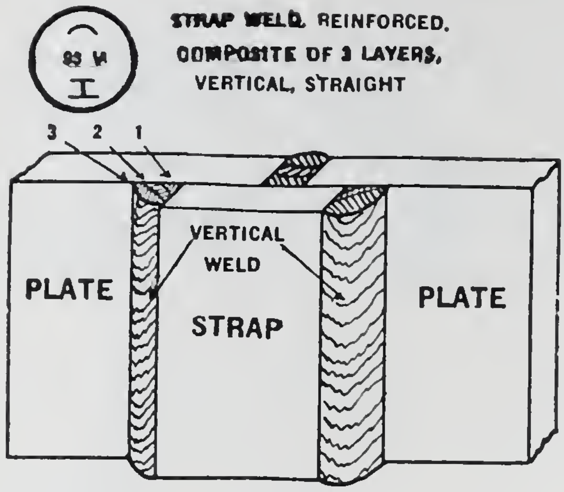


Flush is a term applied to a weld when the top layer is finished perfectly flat or on the same plane as on the adjoining material as shown at D and E above; or at an angle of 45 degrees when used to connect two surfaces at an angle to each other, as at F above. This type of weld is to be used where a maximum tensile strength is not all important, and must be specified by the designer—together with a minimum number of layers of welding material.

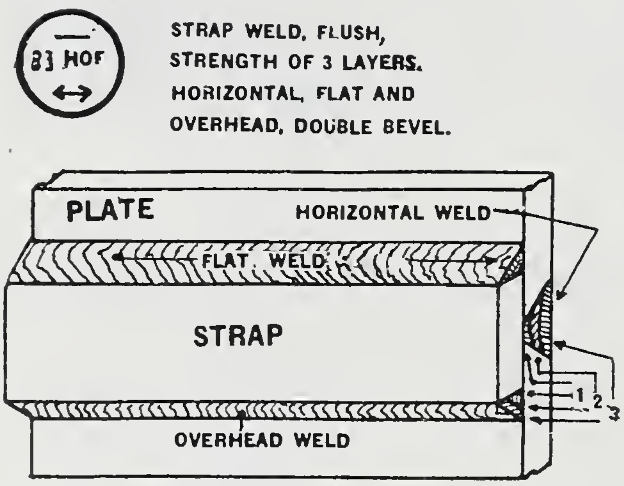


Concave is a term applied to a weld when the top layer finishes below the plane of the surrounding material, as at G above; or beneath a plane of 45 degrees at an angular connection, as at H and J above. To be used as a weld of no further importance than filling in a seam or opening, or for strictly calking purposes when it is found that a minimum amount of welding material will suffice to sustain a specified pound square inch pressure without leakage. In this type of weld it will not be necessary for the designer ordinarily to specify the number of layers of material, owing to the lack of structural importance.

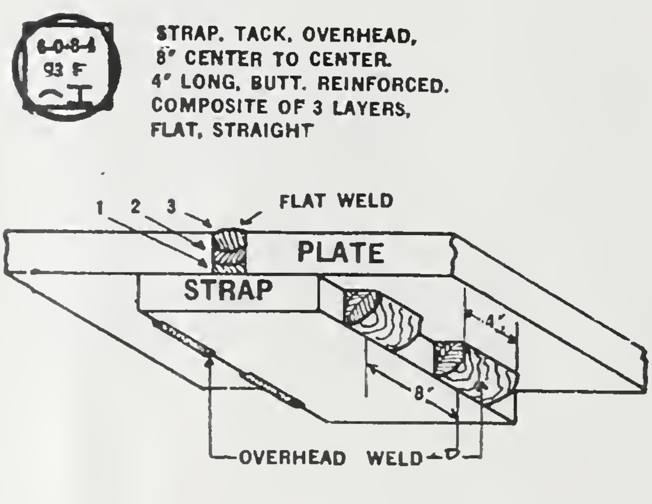
COMBINATIONS OF SYMBOLS



This sketch and symbol show a strap holding together two plates, set vertically, with the welding material applied in not less than three layers at each edge of the strap, as well as between the plates with a reinforced composite finish, so as to make the welded seams absolutely water-, air-, or oil-tight, and to attain the maximum tensile strength. The edges of the strap and the plates are left in a natural or sheared finish. This type of welding is used for the most particular kind of work where maximum strains are to be sustained.

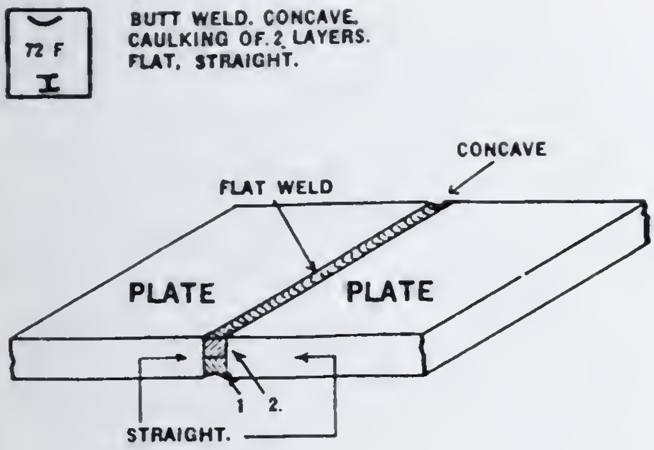


This illustration shows a strap holding two plates together horizontally, welded as a strength member with a minimum of three layers and a flush finish. Inasmuch as the strap necessitates welding of the plates from one side only, both edges of the plates are beveled to an angle, the degrees of which are left to the discretion of the designer. The edges of the strap are left in a natural or sheared state, and the maximum strength is attained by the mode of applying the welding material, and through the sectional area per square inch exceeding the sectional area of the surrounding material.

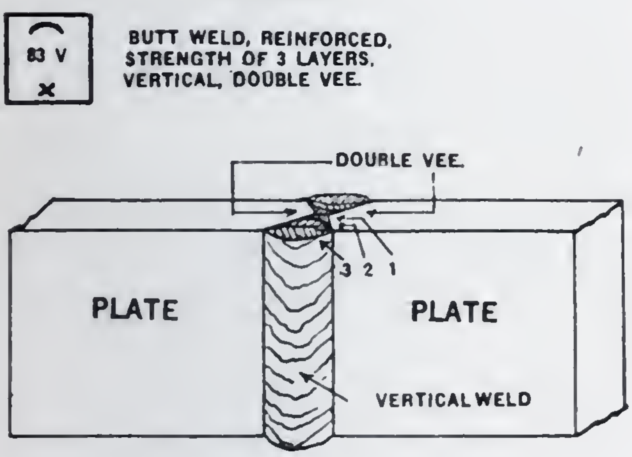


This symbol represents two plates butted together and welded flat, with a composite weld of not less than three layers, and a reinforced finish. A strap is attached by means of overhead tacking, the tacks being four inches long and spaced eight inches from center to center. In this case, the welding of the plates is of maximum strength and water-, air-, or oil-tight, but the tacking is either for the purpose of holding the strap in place until it may be continuously welded, or because strength is not essential. All the edges are left in their natural or sheared state.

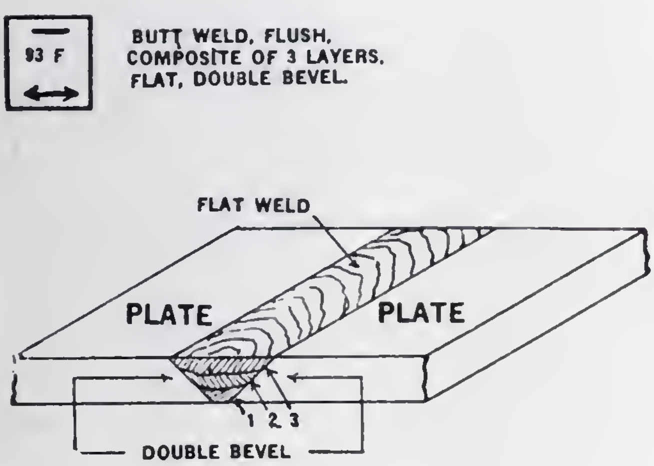
COMBINATIONS OF SYMBOLS



The symbol shown represents a butt weld between two plates, with the welding material finished concave and applied in a minimum of two layers to take the place of calking. The edges of the plates are left in a natural shear-cut finish. This symbol will be quite frequently used for deck plating or any other place where strength is not essential, but where the material must be water-, air-, or oil-tight.

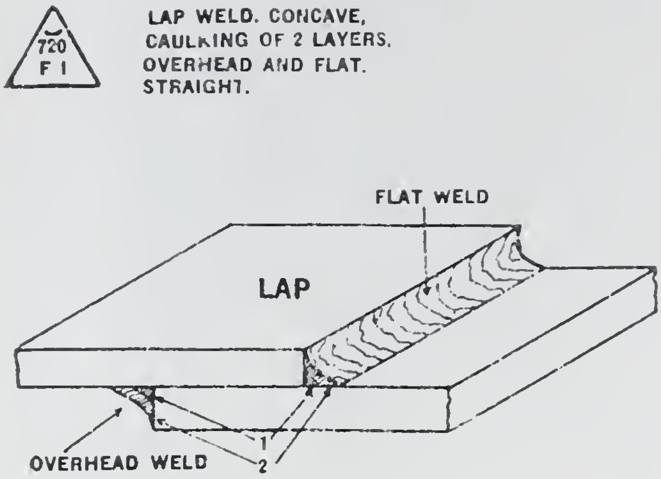


This symbol is used where the edges of two plates are vertically butted together and welded as a strength member. The edges of the adjoining plates are finished with a "double vee," and the minimum of three layers of welding material applied from each side, finished with a convex surface; thereby making the sectional area per square inch of the weld greater than that of the plates. This will be a conventional symbol for shell-plating or any other members requiring a maximum tensile strength, where the welding can be done from both sides of the work.

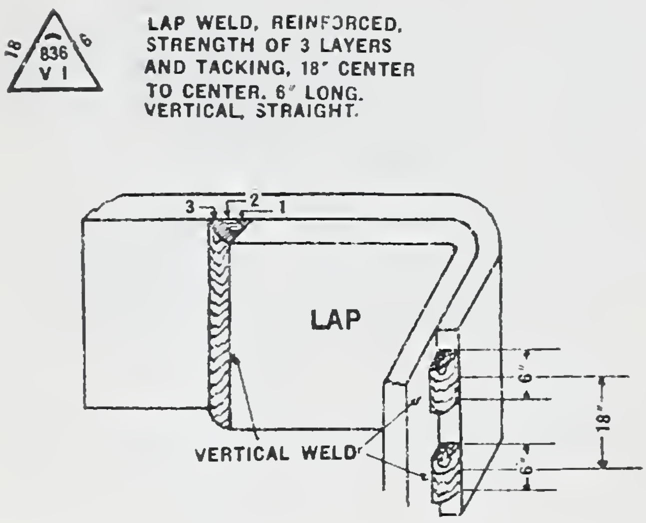


This symbol represents two plates butted together in a flat position where the welding can be applied only from the top surface. It shows a weld required for plating where both strength and water-tightness are to be considered. The welding material is applied in a minimum of three layers and finished flush with the level of the plates. Both edges of the adjoining plates are beveled to an angle, the degrees of which are left to the discretion and judgment of the designer. It should be used only when it is impossible to weld from both sides of the work.

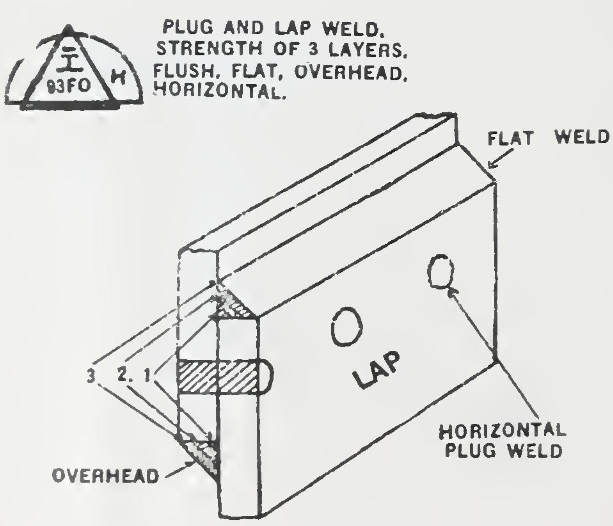
COMBINATIONS OF SYMBOLS



The sketch shows the edges of two plates lapping each other, with the welding material applied in not less than two layers at each edge, with a concave calking finish, so applied as to make the welded seams absolutely water-, air-, or oil-tight. The edges of the plates themselves are left in a natural or sheared finish. Conditions of this kind will often occur around bulkhead door-frames where maximum strength is not absolutely essential.

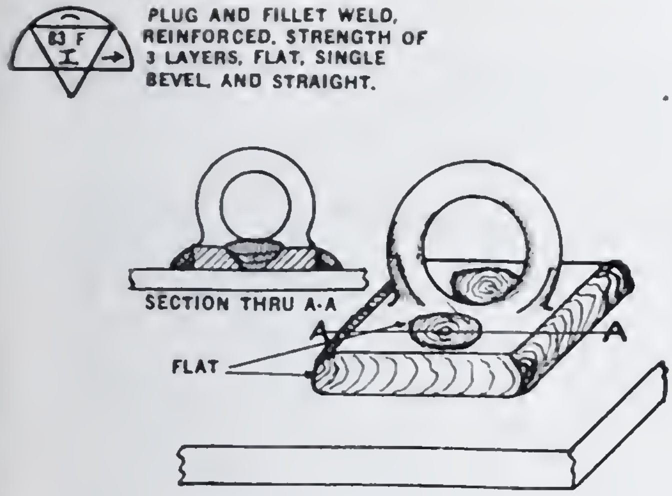


The illustration here shown, is somewhat exaggerated as regards the bending of the plates, but it is shown this way only to illustrate fully the tack and continuous weld. It shows the edges of the plates lapped, with one edge welded with a continuous weld of a minimum of three layers with a reinforced finish—thereby giving a maximum tensile strength to the weld—and the other edge of the plate, tack welded. The tacks are 6 inches long with a space of 12 inches between the welds, or 18 inches from center to center of welds. In both cases, the edges of plates are left in a natural or sheared state.

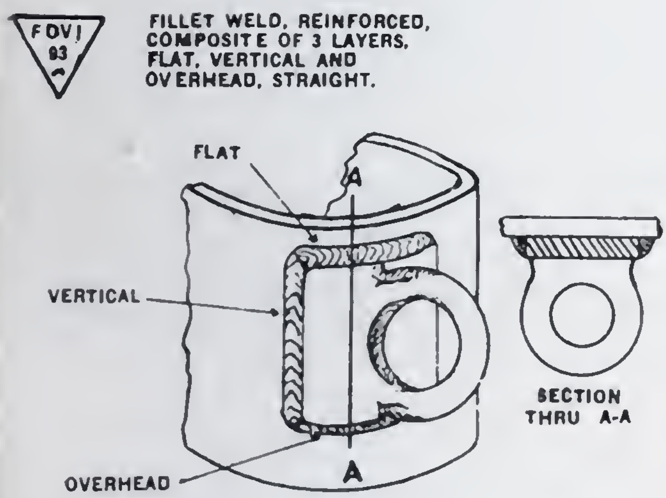


The sketch shows a condition, exaggerated, which is apt to occur in side plating where the plates were held in position with bolts for the purpose of alignment before being welded. The edges are to be welded with a minimum of three layers of welding material for a strength weld, and finished flush; and, after the bolts are removed, the holes thus left are to be filled in with welding material in a manner prescribed for strength welding. The edges of the plates are to be left in a natural or sheared state, which is customary in most cases of lapped welding.

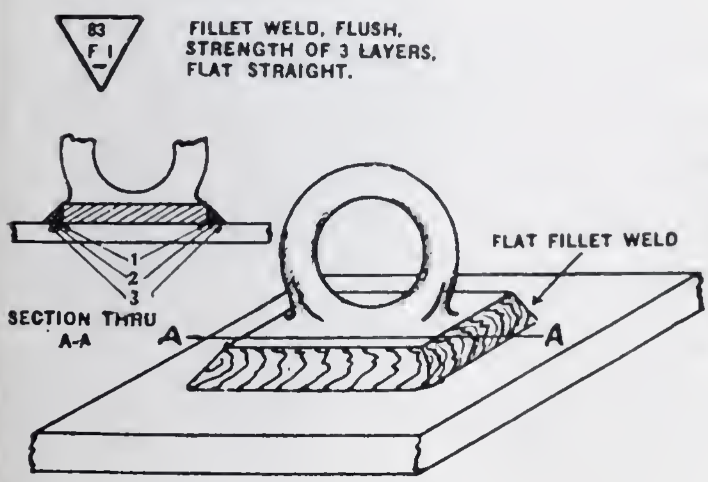
COMBINATIONS OF SYMBOLS



The adjoining sketch shows a pad eye attached to a plate by means of a fillet weld along the edge of the fixture, and further strengthened by plug welds in two countersunk holes drilled in the fixture. The welding material is applied in a flat position for a strength weld with a minimum of three layers and a reinforced finish. The edges of the holes are beveled to an angle, which is left to the judgment of the designer, but the edges of the fixture are left in their natural state. This method is used in fastening fixtures, clips, or accessories that would be subjected to an excessive strain or vibration.

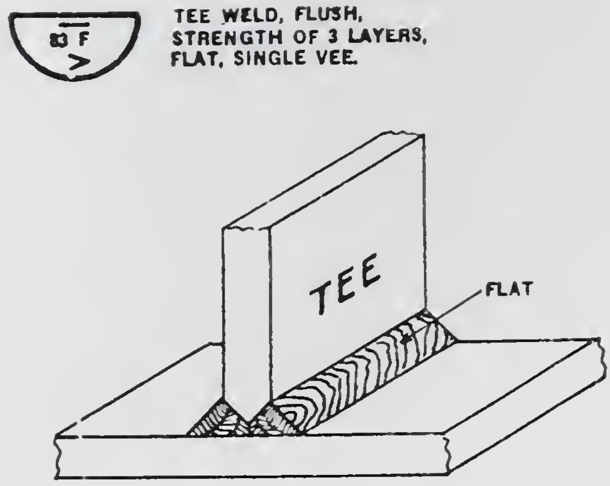


This illustration shows a fixture attached to a plate by means of a composite weld of not less than three layers with a reinforced finish. The fixture, being placed vertically, necessitates a combination of flat, vertical, and overhead welding in the course of its erection. Although a fixture of this kind would never be required to be water-tight, the composite symbol is simply as a possibility of a combination.

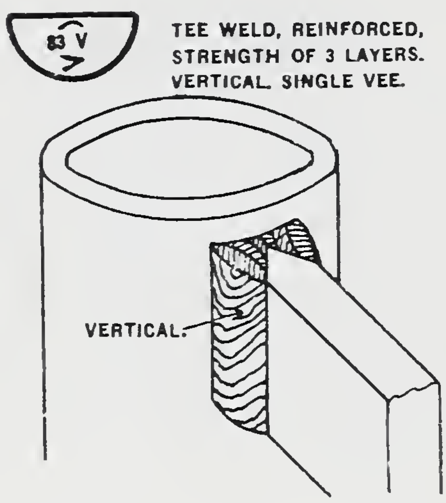


This symbol represents a fixture attached to a plate by a strength fillet weld of not less than three layers, finished flush. The edges of the fixture are left in their natural state, and the welding material applied in the corner formed by the vertical edge of the fixture in contact with the face of the plate.

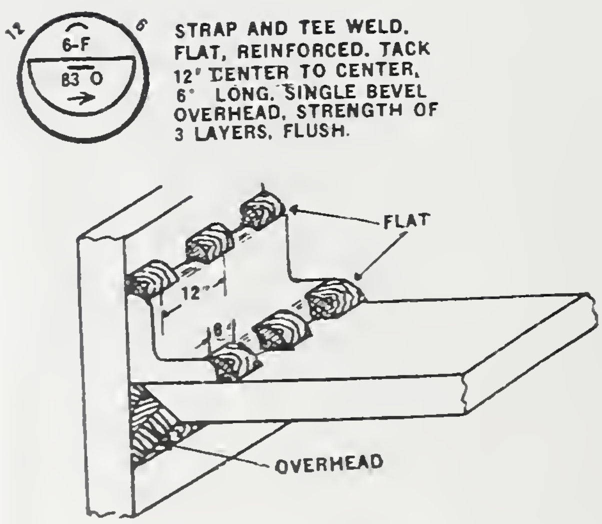
COMBINATIONS OF SYMBOLS



The adjoining sketch illustrates the edge of a plate welded to the face of another plate, as in the case of the bottom of a transverse bulkhead being welded against the deck plating. To obtain a maximum tensile strength at the joint, the edge of the plate is cut to a "single vee," and welded on both sides with a strength weld of not less than three layers, and finished flush. This would be a convenient way of fastening the intercostals to the keelsons. In this particular case, the welding is done in a flat position.



This symbol shows another case of tee weld, with the seam in a vertical position and the welding material applied from both sides of the work. The edge of the plate is finished with a "single vee," and a minimum of three layers of welding material is applied from each side; finished with a convex surface, thereby making the sectional area per square inch of the weld greater than that of the plate, allowing for a maximum tensile strength in the weld.



The illustration herein shown, represents an example of the possible combination of symbols. An angle-iron is tack welded to the plate in the form of a strap or stiffener though in actual practice this might never occur. The tacks are spaced 12 inches from center to center, and are 6 inches long, and applied in a flat position with a reinforced finish. As the strap prevents welding the plate from both sides, the edge of the plate is beveled, and the welding material applied for strength in not less than three layers in an overhead position, and finished flush. Note that in specifying tack welds, it is essential to give the space from center to center of weld, and length of weld by use of figures representing inches placed either side of the circumscribing symbol of the combination.

DISCUSSION

Mr. C. D. Terry:* I should like to ask the speaker of the evening a question as to the preparation of the weld—that is, the preparation of the edges of the metal to be welded. I believe with a butt-weld it would be necessary to double bevel the edges. Is that true, and is there any other preparation necessary?

Mr. H. A. Hornor: On thin material it is not necessary, but it becomes necessary on very thick material. Unfortunately there is one question regarding beveling which the Research Committee has never answered—that is, the question of the degree of angle of the bevel. There is a difference of opinion, and further investigation will probably have to be made before that is settled. The 45-degree angle is the one ordinarily used, but 60 degrees can be used. The greater the bevel the more electrode used.

I did not touch on one question that I thought would come up in the discussion—that is, the type of joint. Lloyd's Register has settled that for England. You noticed in all the designs I showed you they lap the plates. That was due to the investigations of Lloyd's on alternating stresses. The weld is safe up to a point of about 6½ tons. That is almost as good as wrought-iron but not as good as mild steel. For that reason Lloyd's Register suggests only a small-sized ship at the present time. They take the point of view that because of the alternating stresses the lap-joint is the preferable type, and the reason is that with a lap-joint you will be able to distribute these stresses back to the plate itself.

With regard to alternating stresses, if you make a ship strong enough there should not be any question about it. There does not seem to be very much doubt about electric welding. Mr. Holslag and myself still contend that if you can make a joint as strong as or stronger than a riveted joint you need not bother about alternating stresses or fatigue or anything of that kind.

*Assistant General Superintendent, National Tube Co., Pittsburgh.

If a riveted joint will open its calking and then finally break at 110 tons, while a welded joint can be made so that there is no movement at the joint, there can hardly be any danger about it. However, Lloyd's are very conservative and they still contend that we must have both a lap-joint and a strap joint. Here is another of their very interesting tests. They riveted a bracket on a plate then proceeded to put a downward pressure on both the riveted bracket and a similar sample of a welded bracket. The riveted joint gave way at something quite insignificant, in the neighborhood of 25 000 or 30 000 pounds per square inch. The welded sample went up to the limit of the machine—70 000 pounds—and they could not move it.

The alternating-stress tests were on a machine that functioned up to 3 000 000 to 5 000 000 alternations, and the welded joint showed weakness when an alternating stress of 6½ tons was applied. This was less than wrought-iron, or mild steel. Mr. Cox, the Assistant Chief Supervisor of Lloyd's in the United States, does not hesitate to approve the recommendation for the building of at least a 5000-ton ship.

Mr. C. J. Holslag:* In welding the main point is the operator, as Mr. Hornor pointed out. Anything that will help the operator out is a step in the right direction. In welding with the metal pencil—metallic electrode welding—you draw an arc with the metal and it immediately melts a little bowl in that metal and the metallic electrode starts passing down into that bowl. If the electrode metal falls outside of that bowl, or if you move so fast you get ahead of the puddle, no weld occurs. The electrode metal may be melted but the plate is not melted and no weld occurs. The ideal condition to get a good weld is the short arc. There is less chance of the electrode metal falling on a wrong spot and there is less chance of the air getting at it and cooling and oxidizing it. And then when you get a long arc the metal falls in globules, and with a very small arc the metal passes over as a vapor and there is no chance but what it will fall into the spot that has become fluid on the work.

*Chief Engineer, Electric Arc Cutting and Welding Co., Newark, N. J.

There are all sorts of electric welding machines, from the water barrel resistance to the freak transformers that our company sells. With a direct-current machine it is very easy to hold a long arc. Your good operator holds a short arc. Much of the discredit attached to electric welding has been due to the long arc. In direct-current operation, much has been done toward shortening the arc by voltage control. If you limit the voltage by relay or by the original voltage of the machine so that a long arc cannot be held, you get this short arc advantage, but you generally decrease the depth to which the arc will penetrate. But with alternating current you can limit that arc through the natural tendency of the alternating-current arc to go out without losing voltage. Over-voltage circuit is sometimes necessary. The voltage of the plain wire arc is from 15 to 22, fluxed wire 20 to 30, slag covered 30 to 40, and carbon arc 50 to 60 volts. The voltage of the arc is low but you need more voltage than that to maintain the arc and penetrate the metal. For covered wire you need more yet to penetrate and blow away the slag. The alternating current is the only method that has yet developed which will give you those three fundamentals—short arc, high open circuit voltage for puncturing, and the use of all sorts of electrodes. But you are still dependent on the operator and the possibility that he does the right thing and puts the metal where it belongs. We claim that with alternating current, he welds wherever he holds the arc.

Mr. M. P. Clark:* I would like to ask Mr. Hornor why it is that the welded joint is as strong under tension as the plate; and yet, if dropped to the floor, it would break on account of being brittle. It seems to me that electric welding is unsatisfactory as a result of the great heat developed. The cooling medium —the surrounding atmosphere—cannot help but cool the joint too quickly and this leaves the metal brittle. I think if the joint, after being made, were cooled slowly, or in other words annealed, it would be more satisfactory.

Mr. H. A. Hornor: When the welded joint is as strong as the plate, the plate material breaks far from the weld. There is no effect on the structure of the steel a sixteenth of an inch

*Engineer Expeditor, National Tube Co., Pittsburgh.

away from the weld. Annealing does not seem to strengthen electric arc welds but rather tends to weaken them; according to recent investigations of some metallurgists, although I am of the opinion that this is a disputed question.

The weld might not be as strong as the plate, but still strong enough to convince you that it had positively good adhesive powers. We have had many tests where the tensile strength tested as high as 45 000 pounds per square inch, which is better than a riveted joint, but it had no elasticity and no ductility. The question of ductility brings us right into line with the dispute between the two types of electrode, the bare and covered, because the covered electrode shows a marked increase in ductility over the bare. That is really the only way ductility has been obtained. The bare wire electrode seems conducive to brittleness.

Mr. C. D. Terry: I should like to ask the approximate size of the tank which was tested under pressure and under vacuum.

Mr. H. A. Hornor: It was 12 by 12 by 12 feet.

Mr. A. B. Holcomb:* From one or two different sources I have been informed that the deposit of metal in the alternating-current arc is more apt to be porous than that deposited in the direct-current arc.

Mr. C. J. Holslag: Mr. Esholz, of the Company Mr. Condy represents, has just gone on record with the statement that porosity comes only from a long arc, and the General Electric Comany has injected into a recent publication the statement that it is just the opposite—that alternating current offers a greater chance of getting rid of porosity because of its short arc. We have made test plates with alternating current that fully equalled anything that was ever done with direct current. A fellow in the West wrote me asking how he could get any metal on the plate if it is jumping back and forth with the alternating current. This idea of the metal following the polarity of the juice seems to be erroneous. You can work up the side and overhead with the

*Standard Tin Plate Co., Canonsburg, Pa.

alternating current, showing that the overhead welding is no function of the positive or the negative electrode. I believe that the explanation is capillary attraction. One drop of water will stay on the ceiling, but two will drop. So if you get the puddle just viscous enough so that the second drop will not fall, you get a weld. And it has nothing to do with either positive or negative current. The advantages of alternating current are obvious, and I believe we have proved it can be applied. Up until recently, direct current was the only way of welding. With direct current you have a lot of moving parts, considerable weight and general maintenance charges; with alternating current you have simply a transformer. With a single change it is much more efficient than a direct-current apparatus with two or three or four changes and moving parts.

MR. W. M. JUDD, *Chairman*:* Mr. Hunter, you are interested in this sort of thing. Have you anything to add?

MR. P. E. HUNTER:† I have listened to the discussion with a great deal of interest, but I am here after information.

MR. A. B. HOLCOMB: In alternating-current welding is there an external reactance in the transformer circuit; and, if so, is it in the transformer itself? I should like to know, also, with what range of frequencies you have experimented.

MR. C. J. HOLSLAG: As to the range of frequencies, we have made successful installations of 25, 30, 33, 40, 50, 60, and 66, and that is all we have been asked for. I think we could meet any frequency. Some of our best plates have been made at 25 cycles.

In regard to external reactance, reactance is not what makes the apparatus operate, but the fact that the characteristics are such that the machine follows the voltage of the arc. You can short circuit any two of the terminals and get but slightly more current than when welding. This is accomplished in our machine

*Member, W. G. Wilkins Co., Pittsburgh.

†President, Independent Bridge Co., Beaver, Pa.

by the secondary being split up differentially so that all these requirements are met instantaneously and automatically. The voltage of the arc consists of three parts; there is an essential voltage—about 10 or 11—below which the arc cannot be held. Then there is the *I R* drop of two or three volts, and the main reason the alternating current must have some special apparatus to hold it is that there is a change in resistance due to the man's hand wavering, imperfections in the metal, dirt or oil, slag, etc. The main variation is caused by the movement of the hand that is holding the arc, and it is necessary to supply, instantaneously, extra voltage which will rise above 22 for bare wire and adjust itself to meet the conditions of the arc at all times.

Mr. P. E. Hunter: Did the General Electric people succeed in welding one-inch plates?

Mr. H. A. Hornor: Yes.

Mr. P. E. Hunter: No longer than 60 days ago I received a letter from the General Electric people stating that they considered this impossible.

Mr. H. A. Hornor: I do not doubt it at all. It is a large organization and it is hard to keep in touch with all its developments. Last week I had my assistant examine the stationary welding machine, and that machine is capable of putting in two spots on two ¾-inch plates. That is its rating, and it will be capable of welding two 1-inch plates. The machine I was speaking of was a special experimental machine built by the General Electric people since the opening of the war. They built that machine extremely heavy to see just how far they could go, and they went up to three thicknesses of 1-inch plate. These machines are the first ones ever built and they are extremely difficult to build. We now have the two small ones, 12-inch and 27-inch, which are capable of ½- to ⅝-inch plate work, but the big machine has not been put in practical operation at the present time.

The Engineers' Society

of

Western Pennsylvania

INCORPORATED 1880

ABSTRACT OF MINUTES

Volume 34

FEBRUARY 1918—JANUARY 1919

ANNUAL MEETING

The Thirty-eighth Annual Meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Pittsburgh, Tuesday, January 15, 1918, at 8:20 P. M., President Alex L. Hoerr presiding, 97 members and visitors being present.

The minutes of the last Annual Meeting held January 16, 1917, were read and approved.

The annual report of the Board of Direction, which included the reports of the Standing and Special Committees, the Sections and the Treasurer, was read as follows:

REPORT OF BOARD OF DIRECTION

The Board of Direction of the Society held eleven regular monthly and three special meetings during the past year, at which routine of business of the Society was transacted.

During the year there were eight regular monthly meetings, one special meeting and the Annual meeting of the Society. Total attendance was 1376, the average being 125. The maximum attendance was 515 at the February meeting and the minimum 60 at the October meeting. The average number participating in the discussion of papers was 7.

At the close of the year the membership of the Society was as follows:

Honoraary Members	3
Members	996
Associate Members	43
Associates	30
Juniors	90
Student Juniors	25
Total	1187
Resignations	32
Removed by death	9
Total	41
Accessions	94

REPORT OF HOUSE COMMITTEE

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

Referring to the matter of work of the House Committee during the past year, would advise that the rooms have been kept in good condition

and the necessary alterations, due to moving from the Oliver Building to the present quarters of the Union Arcade Building, have been made.

The fire insurance on the Society's property was considered too low in amount and this has been doubled, bringing it to $2000.00. The chairs and tables have been refinished so as to present a better appearance.

Two matters that have been the subject of considerable discussion for some years past have been taken up and made effective, namely the procuring of some popular literature for the Reading Room, which seems to be appreciated, and the amount of this literature could probably be increased to advantage. The other change that has been made consists in keeping the rooms open in the evenings. They are now open uutil 10:30 at night, and, while the evening attendance is not as great as might be expected, nevertheless, it is sufficient to warrant the continuance of this practice as it seems likely that the use of the rooms in the evenings will increase.

Respectfully submitted,

GEORGE H. DANFORTH, *Chairman House Committee.*

REPORT OF PUBLICATION COMMITTEE

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

During the year four meetings of the Committee were held, with an average attendance of seven.

Papers read at the general meetings of the Society.... 8
Papers read at the meetings of the various Sections.. 10
Two lecture presented.

Of these eighteen papers have been published in the Proceedings of the Society or will appear in later issues.

In November Mr. E. H. McClelland, Technology Librarian, Pittsburgh Carnegie Library, was engaged to edit the Proceedings and now has this work in charge.

Arrangements have been made to issue a bi-monthly bulletin to be added to the notices of meetings sent out. These bulletins will contain personal and news items, abstracts of committee reports, digest of minutes of the Board of Direction and such other items as will be of special interest to the members of the Society.

It is believed that, with sufficient cooperation on the part of the members in the way of contributing news items, such a bulletin will prove to be very interesting to the members.

ROBERT LINTON, *Chairman, Publication Committee.*

REPORT OF ENTERTAINMENT COMMITTEE

To the Engineers' Society of Western Pennsylvania:

During the past year your committee arranged the following inspection trips and entertainments:

INSPECTION TRIPS

November 17—To the Plant of the American Steel & Wire Co., Donora, Pa., to inspect their new Industrial Housing Development near Donora and their new Duplexing Dept.

SMOKERS

January 20th—Held at Fort Pitt Hotel.
October 6th—Patriotic Smoker held at the Fort Pitt Hotel.
December 7th—Smoker at Fort Pitt Hotel.

PRESENTATION OF REGIMENTAL COLORS TO 5TH U. S. ENGINEERS

July 2nd—U. S. Flag was presented to the 5th U. S. Engineers at their Camp near Oakmont, Pa. The presentation was made by Dr. John A. Brashear and the colors received by Col. Jadwin. The regimental flag was later forwarded to Col. Jadwin at Oakmont and lost in transit. It was located and eventually returned to the manufacturer, who changed the lettering to "15th U. S. Engineers" which is the present designation of the regiment and forwarded to Col. Jadwin in France, December 11th.

HOUSE WARMING

June 15th—A House Warming was held on removal to our new quarters in the Union Arcade Building.

The Annual Banquet was not held this year on account of war conditions.

Respectfully submitted,

GEORGE H. NEILSON, *Chairman.*

REPORT OF MEMBERSHIP COMMITTEE

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

I wish to submit the following report for the Membership Committee for the year 1917. During the year 113 new members were added to the

Society. During the same period, 48 members have been lost by resignation and death, as follows:

By resignation	35
By death	12
Dropped	1
Total	48

which leaves a net increase of 65 members added to the Society during the past year.

The Membership Committee has experienced difficulty in interesting new members in the Society at the present time. Since war was declared, many men have joined the military organization and many of the others are so actively engaged in furnishing supplies, either directly or indirectly to the military organization, that it is difficult to secure their interest at this time.

The Society is doing a splendid work and should receive the support of all engineers in this vicinity, particularly at this time, when cooperation in engineering work is so essential. The Chairman, therefore, urges the members to use every effort possible to assist the Committee in maintaining our membership up to its present standard during these very trying times.

Respectfully submitted,

H. D. James, *Chairman Membership Committee.*

REPORT OF FINANCE COMMITTEE

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

Your Finance Committee beg to submit the following report of work done during the year 1917.

Monthly audits have been made of the Secretary's Financial Statements and were always found in good order.

In July, all books and accounts of the Society to July 1, 1917, were audited by Ernst & Ernst, Certified Public Accountants, we beg to quote the following from their report:

"During the course of our examination we were very favorably impressed with the neatness and accuracy of your accounting records and with the completeness of same."

During the year, the Secretary and Treasurer of the Society were bonded for $5000.00 each, which was the amount suggested by the Bonding Company after they had investigated the matter.

George H. Barbour, *Chairman.*

REPORT OF COMMITTEE ON ONE HUNDRED FOOT STANDARD

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

Your Committee on One Hundred Foot Standard would report that the room in the basement of the new City-County Building has been plastered, painted, etc., and is ready for use.

The electric lighting system to meet our special requirements has been planned by Mr. S. G. Hibben and will be installed shortly. Wall brackets to support the bench are now being manufactured according to details designed by Mr. Stupakoff. Mr. Stupakoff is working on the details as rapidly as is required. Progress in this installation has been rather slow, but satisfactory.

Respectfully submitted,

Louis P. Blum,

J. G. Chalfant,

S. M. Stupakoff,

N. S. Sprague,

Committee.

REPORT OF TREASURER

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

Your Treasurer takes pleasure in presenting the following statement of the finances of the Society for the year ending Dec. 31, 1917:

INVESTMENTS

Building Fund

One	$1000 Butler Water Company 5 percent Bond No. 9 matures September 2, 1931	$ 1 025.00

Permanent Fund

Two	$1000 Connellsville Water Co., 5 percent Bonds Nos. 317-317, maturing Oct. 1, 1930	2 020.00
Two	$1000 Portsmouth, Berkley & Suffolk Water Co., 5 percent Bonds, Nos. 465-466, maturing November 1, 1914	2 000.00

Two	$1000 Jamison Coal & Coke Co., 5 percent Bonds Nos. 1502-1503, maturing November 1, 1931	2 000.00
Two	$1000 Union Steel Co., 5 percent Bonds, Nos. 36642-36643, maturing December 1, 1952	2 090.00
Two	$1000 Pennsylvania Railroad Co., 4½ percent Bonds, Nos. 27320-27321, maturing August 1, 1960	2 070.00
Three	$1000 Jones & Laughlin Steel Co., 5 percent Bonds, Nos. 3020-3021-3022, maturing May 1, 1931	2 997.92
	Total fourteen Bonds	$14 202.92

Receipts

Dues 1918	$ 25.63
Dues 1917	10 069.05
Dues 1916	615.44
Dues 1915	163.25
Dues 1914	104.36
Dues 1913	10.00
Dues 1911	15.00
Total Dues	$11 002.73
Entrance Fees	890.00
Advertising	1 338.79
Proceedings	452.50
Pins	75.09
Rent of Auditorium	122.50
Banquet Receipts	360.00
Smoker Receipts	675.50
House Warming Receipts	14.00
Interest	742.88
Miscellaneous	232.26
Total Receipts for 1917	$15 906.25

Expenditures

Administration	$ 4 218.69
Entertainment Committee	1 008.52
House	3 918.00
Library	23.85
General Society	1 725.46
Mechanical Section	168.75
Structural Section	185.72
Metallurgical & Mining Section	105.27
Proceedings	2 420.64

Membership Committee	57.50
Miscellaneous	332.65
Total Expenditures for 1917	$14 165.05

Some of the above bonds fluctuate in their market value and two or three are at present well below par. However, since they pay full interest on their face value, and since it is hoped that their market value will soon recuperate, it is justifiable to enter them in the following table at par:

ASSETS

	Dec. 31, 1916	Dec. 31, 1917
Building Fund		
Bond	$ 1 025.00	$ 1 025.00
Cash (Fidelity Title & Trust Co.)	586.37	586.37
Permanent Fund		
Bonds	13 177.92	13 177.92
Cash (Fidelity Title & Trust Co.)	0.00	473.34
Reserve Fund		
Cash (Fidelity Title & Trust Co.)	588.34	2 500.00
General Fund		
Cash (Diamond National Bank)	955.22	311.42
	$16 332.85	$18 074.05
Increase during 1917	1 741.20	
	$18 074.05	$18 074.05

The receipts in 1916 were $22 971.21, or $7 064.96 larger than in 1917 as given above, and if not explained would be discouraging. In 1916 we had three important items entering into the receipts as well as the expenditure columns, which we did not have in the last year: namely the receipts from Banquets held in 1916, the receipts from military engineering lectures and the receipts from the A. S. C. E. Smoker amounted to $8 335.01 and were balanced by proportionate expenditures.

The dues received in 1916 were $10 260.11, or $742.62 less than in the past year.

The entrance fees in 1916 were $780.00, or $110.00 less than during the past year.

Some time ago the Reserve Fund was created to bridge over certain periods when there is actually no income to the Society. This is the first time that this fund has been paid in full.

Likewise has the Permanent Fund for the first time in many years been credited with all the moneys assigned to it by the By-Laws of the Society.

Respectfully submitted,

A. Stucki, *Treasurer.*

REPORT OF MECHANICAL SECTION

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

Four regular meetings of the Mechanical Section were held in 1917, with an average attendance of 70. The maximum attendance was 89 at the October meeting and the minimum 70 at the December meeting. The average number participating in the discussion of papers was 7.

The papers presented were as follows:

February Meeting: "The Burning of Blast Furnace Coke Under Boilers," a Topical Discussion.

April Meeting: "Cotton Rope for Power Transmission," by J. Melville Alison, Engr., William Kenyon & Sons, Ltd., Dukinfield, England.

October Meeting: "Modern Condenser Practice," by D. D. Pendleton, Dist. Sales Manager, Wheeler Condenser & Engineering Co., Pittsburgh, Pa.

December Meeting: "The Manufacture and Use of Die Castings," by Charles Pack, Chief Chemist and Metallurgist, Doehler Die Casting Co., Brooklyn, N. Y.

Respectfully submitted,

ALBERT KINGSBURG, *Vice Chairman.*

REPORT OF STRUCTURAL SECTION

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

I beg to submit report of work done by the Structural Section during the year 1917.

Four regular meetings of the Section were held during the year. The average attendance at these meetings was 88; the maximum being 135 at the November meeting and the minimum 60 at the May meeting.

An average of ten participated in the discussion of the several papers presented. The papers presented during the year were as follows:

January Meeting: "Historic Failures of Masonry Structures," by Prof. Horace R. Thayer, Asst. Professor Structural Design, Carnegie Institute of Technology, Pittsburgh.

March Meeting: "Fireproofing and Fireproofing Methods," read by J. L. Shields, Engr., National Fireproofing Co., in absence of author, Mr. J. P. Barr.

May Meeting: "Some Fundamentals of School House Design," by Mr. C. L. Wooldridge, Superintendent Buildings, Board of Public Education, Pittsburgh.

November Meeting: "Industrial Housing," by H. Walter Forster, General Manager, Independence Bureau, Philadelphia, Pa.

Respectfully submitted,

GEORGE W. NICHOLS, *Chairman*.

REPORT OF METALLURGICAL & MINING SECTION

To the Board of Direction,

Engineers' Society of Western Pennsylvania:

Two regular meetings of the Metallurgical & Mining Section of the Society were held in 1917, with an average attendance of 81, the maximum attendance being 117 at the January meeting, and the minimum 45 at the November meeting.

The average number participating in the discussion of the papers was 11.

The papers presented were:

January Meeting: "The Possibilities of Smokeless Operation of Heating Furnaces and Soaking Pits,"—A Topical Discussion.

November Meeting: "The Situation Regarding Manganese, Pyrite, Sulphur, Chromium and Sulphur," by J. E. Johnson, Jr., Consulting Engineer, New York City.

Respectfully submitted,

W. L. AFFELDER, *Chairman*.

REPORT OF TELLERS

To the Members of the Engineers' Society of Western Pennsylvania:

The undersigned Tellers publicly canvassed the ballots in the Annual election of officers of the Society at noon, Tuesday, January 15, 1918, and beg to report the following results:

Ballots received	215
Irregular ballots	22
Ballots counted	237
For President—W. E. Snyder	215
For Vice-President—W. C. Hawley	210
For Treasurer—A. Stucki	214
For Directors	
A. N. Diehl	207
Robert Linton	200

Respectfully submitted,

E. D. LELAND,
JOHN A. HUNTER,
WM. HOOPER,
Tellers.

The President thereupon declared the following men elected:

For President....................................W. E. Snyder
For Vice-President..............................W. C. Hawley
For Treasurer...A. Stucki
For Directors..
....A. N. Diehl
....Robert Linton

Before asking the President elect to take the chair, Mr. Hoerr gave a short talk:

"I would like to say a word and thank the members of the Society for the opportunity that was given me to serve them during the past year and to thank the members of the Board of Direction and the officers for their assistance and cooperation. This year, of course, has been a very eventful one. There have been many other things to do than operate an Engineers' Society and the temptation was frequently strong to let it lag. It is undoubtedly necessary, however, to continue the Society as strongly and vigorously as possible, because only by the most thorough assistance from all engineers can the proper war preparations be made. I would, therefore, ask all of you in this coming year to continue and if possible to make an increase in your interest in the Society, and in the help you will give the incoming officers.

"The President requested the two Junior Past Presidents, Samuel E. Duff and A. Stucki to escort the President-elect—W. E. Snyder—to the chair, who thereupon addressed the Society as follows:

"I want to take this opportunity to make a few remarks, and in order that I may say only what I wish, without waste of words or time, I have written them. Lest any one think this is the preface to a boastful forecast of a new regime, I will quote as a text, the latter part of the 11th verse of the 20th Chapter of I Kings—' . . . let not him that girdeth on his armour boast himself as he that putteth it off . . . ."

"I have been a member of this Society nineteen years. During most of this period, I have taken various parts in its activities. During the past seven years I have served on the Publication, Membership, Finance and Special Committees; as Chairman of the Mechanical Section; as a Director, and as Vice-President. In all of this work, I have formed most pleasant and profitable associations with many men, as officers and members of the Society. I have come to have a close knowledge of the character of the principal work of this Society as indicated by the papers and discussions in its Proceedings. From first hand knowledge, I know there is no camouflage about the men who have given, and are giving, their best in this work of the promotion of engineering; nor is there camouflage about the subject-matter of the Proceedings. Membership and papers, as I know them, give abundant evidence that the first and main work of this Society is the advancement of engineering. It is because of the splendid character of the men actively engaged in this work, and the intrinsic merit of its results that I have been willing to do what I might to help, and I am proud that it falls to my lot to serve for a little time as its President.

I have been a member of other engineering organizations—The A.S. M. E., the American Iron & Steel Institute, the Deutscher Verin Ing., the Eisenhuttenleute. I dropped the two German Societies shortly after the war began, in 1914. If it were ever necessary, I would drop the other

two, before I would give up membership in this one. I could name from eight to twelve different subjects of papers and discussions in the Proceedings of the past five years, which are the only ones of their kind in the whole field of technical literature. The making available for ready reference of busy engineers, of such highly specialized engineering information and data—much of it of especial local interest—is a work of the greatest importance to the advancement of engineering knowledge.

"I have given a great deal of study to the question as to what should be the special activities of this Society, if any, because of the war in which we are now engaged. I am fully convinced that at present there is no reason or occasion for any unusual action of this Society as a body. There are now so many different and well organized lines of work for aiding in the prosecution of the war, that there is abundant opportunity for each individual to do his part in the way best adapted to his circumstances. I refer to this now, because silence or absence of special action at a time when there is so much discussion and action on the part of other organizations, might be misinterpreted as apathy or indifference.

"The work, however, which this Society is peculiarly fitted to do, as its part in the war, it is now doing and has been doing for years past. It is the promotion of the application of engineering in the varied productions of this district. This war has been referred to, not inaptly, as a war of engineers. Certain it is that engineering work is playing a wonderfully important part in its every phase. If we, and our allies, are to win, our engineering must be superior, or at least equal to that of Germany.

"I have had opportunity in the past of making two extended trips in Germany, under peculiarly favorable conditions. They were such that I had entrance into a large number of manufacturing plants where I could study their methods in almost any line I wished, at short range. I have always admired the high grade character of their engineering work, and have emphasized this in various papers and reports over and over again, as well as the large proportion of engineers and other technical men engaged in their manufacturing plants. Developments since then have made plain to the world their willingness to prostitute to a devilish purpose this high technical proficiency. These same developments have also made plain what must be accomplished by the allied engineers to insure German defeat. I have no doubt whatever of the final result, but I know one of the most powerful means of achieving such favorable outcome is the normal and usual work carried on by bodies of engineers such as this.

"Engineering work is necessarily unobstrusive—it vaunteth not itself—it is done in the quest of a room, or in an office over a table or drawing-board. When its results appear, their cause or source is often overlooked, or completely forgotten in the glory of their appearance. In the midst of the rush and bustle incident to the hurried preparation for a great war, it is hard for some individuals and societies to believe that the greatest effect for the common good, is produced by a continuation of their usual normal line of activity. This is especially true condition when that normal work is to aid in the development of the proficiency of the engineer, whose products are all important to the prosecution of the war.

"I believe I neither exaggerate nor distort any fact when I say that the products of this district will play a very important part in the winning of this war and that in the making of these products so necessary now to this Nation, the work of engineers is becoming of increasing importance. That the work of this Society, so well founded and carried forward by our predecessors for 38 years, shall be dedicated anew to the end that it may serve to aid, strengthen and encourage our engineers to

their highest technical achievements, should be our purpose, born of a sincere and earnest patriotism."

The Secretary read a cablegram from Major Hiles as follows:

"Members, Engineers' Society of Western Penna.:

Wishing you all a happy Christmas and a bright New Year. Hope to greet you face to face in the day to come, after our work here is finished."

No further business coming before the Society, a paper on "Cantonment Construction" was presented by Mr. Morris Knowles, Consulting Engineer, Pittsburgh.

On motion the meeting adjourned at 10:22 P. M.

K. F. TRESCHOW, *Secretary.*

STRUCTURAL SECTION

The Annual Meeting of the Structural Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., January 8, 1918, at 8:10 P. M., Chairman George W. Nichols presiding, 103 members and visitors being present.

The minutes of the last Annual Meeting were read and approved.

Mr. J. A. Ferguson on behalf of the Nominating Committee, reported the nomination of the following officers for the ensuing year:

George H. Danforth..Chairman
W. M. Judd..Vice Chairman
Kenneth H. Talbot..
C. W. Bretland..
C. N. Haggart..
....Directors

No further nominations being made, the Secretary was instructed to cast a unanimous ballot for the members named, who were thereupon declared elected

There being no further business, the meeting adjourned at 9:00 P. M.

K. F. TRESCHOW, *Secretary.*

METALLURGILCAL & MINING SECTION

The Annual Meeting of the Metallurgical & Mining Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, January 29, 1918, at 8:14 P. M., Chairman W. L. Affelder presiding, 101 members and visitors being present

The minutes of the last Annual Meeting held January 30, 1917, were read and approved.

The Annual report of the Chairman was read by the Secretary.

The report of the Nominating Committee was presented as follows:

To Officers and Members of Metallurgical & Mining Section

Engineers' Society of Western Pennsylvania:

Dear Sirs:

Your Nominating Committee have selected the following names as their recommendation for officers of the Metallurgical & Mining Section of the Engineers' Society of Western Pennsylvania for the ensuing year:

F. N. Speller........Chairman
Frederick Crabtree........Vice Chairman
Robert Linton
J. W. Paul
C. F. Taylor
A. W. Patton
R. R. Hice........Directors

Respectfully submitted,

W. E. Fohl,

W. McA. Shiras,

J. O. Handy.

On motion nominations were closed and the Secretary was requested to cast a unanimous ballot in favor of the election of the officers named, who were thereupon duly declared elected.

On motion the meeting adjourned at 8:40 P. M.

K. F. Treschow, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg. Tuesday, Feb. 5, 1918 at 4:20 P. M., President W. E. Snyder presiding, Messrs. Hoerr, James, Pittman, Danforth, Kingsbury, Barbour, Stucki, Duff and the Secretary being present.

The minutes of the last regular meeting held Jan. 8, 1918 were read and approved.

The applications of the following gentlemen, having been regularly published to the Society, pursuant to the action of the Board, were duly elected to membership.

MEMBERS

Martin, John Dickerson — Ottinger, Harry
Norris, William H. — Rogers, Frederick Drummond
Shultz, Frank Walter

ASSOCIATE MEMBER

Dodds, Leonidas Louis

JUNIORS

Pugh, George Arthur — Spencer, Herbert Lincoln

The applications of the following gentlemen were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBER

Herr, Benjamin M.

ASSOCIATE MEMBERS

Burd, Francis John — Knapp, James Howard

The Secretary announced that Mr. Willard P. Chandler, Jr., who joined the Society in June, 1914 and resigned December, 1917, desired to be reinstated, whereupon the Secretary was authorized to replace his name upon the Society Rolls.

The Secretary presented letters of resignation from the following gentlemen:

Abbe, Walter, Jr.	Isherwood, John, Jr.
Boyd, David F.	Kuhn, J. I.
Davis, A. L.	Mark, P. C.
Dudley, Wray	Partington, J. A.
Snyder, G. T.	Sanderson, W. W.

which, after discussion, were ordered accepted.

Letters were also received from the following gentlemen and their resignations ordered tabled:

Hitchcock, H. K. Fisher, H. W.
Sloane, D. M.

The Secretary announced the death of Mr. John L. Haines, who joined the Society January 1904 and died December 14, 1917.

Mr. Danforth was requested to ask Mr. W. C. Moreland to prepare a memoir of Mr. Haines for publication in the Proceedings.

The report of the Secretary, showing the financial condition of the Society at the close of business December 31, 1917, having been regularly audited by the Finance Committee, was approved.

No reports were made by the Chairmen of the various Committees, due to the Committees having just been appointed for the year and no work having been taken up as yet.

The Secretary recommended that Miss Harper, stenographer in the Society's office be given an increase of $10.00 per month. The secretary was authorized to make this increase.

The Secretary retired from the meeting while the election of a Secretary for the ensuing year was considered. Mr. K. F. Treschow was re-elected as Secretary with increase in salary of $25.00 per month.

The meeting adjourned at 5:10 P. M.

K. F. Treschow, *Secretary.*

SPECIAL MEETING
BOARD OF DIRECTION

A special meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg. Tuesday, February 5, 1918 at 5:15 P. M., President W. E. Snyder

presiding, Messrs. Hoerr, James, Pittman, Danforth, Kingsbury, Barbour, Stucki, Duff and the Secretary being present.

President Snyder stated that the meeting had been called to consider the advisability of transferring all moneys and the bond now in the Building Fund to the Permanent Fund.

The Secretary stated that in accordance with the request of the Board of Direction, he had endeavored to secure the names of the men contributing the money, but after a thorough search of the records of the Society and inquiring of several men who were members of the Society at the time this fund was created, was unable to obtain any information whatever.

After a general discussion, it was moved and carried that the transfer be recommended to the Society and ballots sent out to all corporate members, together with statement explaining why it was considered advisable to make this transfer. The Chairman of the Finance Committee was requested to write up this statement.

The meeting adjourned at 5:45 P. M.

K. F. Treschow, *Secretary.*

MECHANICAL SECTION

The regular bi-monthly meeting of the Mechanical Section of the Engineers 'Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, Feb. 5, 1918 at 9:00 P. M., Vice Chairman L. C. Frohrieb presiding in the absence of Mr. Kingsbury, Chairman, 21 members and visitors being present.

The Minutes of the last regular meeting held December 4, 1917 were read and approved.

No further business coming before the Section, the paper of the evening on "Electricity as a Substitute for Natural Gas for Heating Purposes" was presented by Mr. Frank Thornton, Jr., Engineer, Electric Heating Dept. Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

Ensuing discussion was participated in by: A. Stucki, Cons. Engr., Pittsburgh, Pa.; H. D. James, Asst. to Mgr. Engineering, Westinghouse Elec. & Mfg. Co.; W. E. Snyder, Mech. Engr., American Steel & Wire Co., Pittsburgh, Pa.; Theo. J. Volkommer, Mech. Engr., Vitro Mfg. Co., Pittsburgh, Pa.; L. C. Frohrieb, Secy., Federal Engineering Co., Pittsburgh, Pa.; H. P. Smith, Power Engr., McClintic Marshall Co., Pittsburgh, Pa.; H. D. Swoboda, Cons. Engr., Pittsburgh, Pa.; T H. Van Aerman, Salesman, General Electric Co., Pittsburgh, Pa.; and the author.

On motion the meeting adjourned at 10:10 P. M.

K. F. Treschow, *Secretary.*

MECHANICAL SECTION—ANNUAL MEETING

The Annual Meeting of the Mechanical Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday evening, February 5, 1918 at 8:15 P. M., Mr. Hoerr presiding, in the absence of the Chairman, 21 members and visitors being present.

The Minutes of the last Annual meeting held Feb. 6, 1917 were read and approved.

The Annual Report of the Chairman was read by the Secretary.

The report of the Nominating Committee was presented as follows:

Officers and Members Mechanical Section,
Engineers' Society of Western Pennsylvania:

Dear Sirs:

The Committee on nominations for the ensuing year begs to report the following members have been nominated:

Chairman, Albert Kingsbury
Vice Chairman, L. C. Frohrieb
Directors: J. C. Hobbs, F. L. Egan, C. W. Littler

Yours very truly,

A. Stucki
A. L. Hoerr
W. Edgar Reed
Nominating Committee.

On motion nominations were closed and the Secretary was instructed to cast a unanimous ballot in favor of the officers named and they were thereupon declared elected.

On motion the meeting adjourned at 9:00 P. M.

K. F. Treschow, *Secretary.*

GENERAL SOCIETY

The 367th regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, February 19, 1918 at 8:05 P. M., President W. E. Snyder presiding, 101 members and visitors being present.

The minutes of the last regular monthly meeting held Dec. 18, 1917 and of the special meeting held Jan. 22, 1918 were read and approved.

The minutes of the February 5th meeting of the Board of Direction were read, announcing the election of five applicants to the grade of Member, one to the grade of Associate Member and two to the grade of Junior and the receipt of three applications for membership. Also the reinstatement to membership of one applicant.

No further business coming before the Society, the paper of the evening on "Producer Gas, Its Manufacture and Use" was presented by Dr. C. S. Palmer, Mellon Institute, University of Pittsburgh .

The ensuing discussion was participated in by: Mr. W. E. Snyder, Mech. Engr., American Steel & Wire Co.; Mr. E. B. Stimpson, Mgr., F. J. McWade Co., Pittsburgh, Pa.; Mr. C. G. Gerber, Engr., Crucible Steel Co. of America, Pittsburgh; Mr. J. C. Carr, Supt. Track Dept. Jones & Laughlin Steel Co., Pittsburgh; Mr. A. Stucki, Cons. Engr., Pittsburgh, Pa.; Mr. R. Hanau, Cons. Engr., Bacharach Industrial Instrument Co.; Mr. J. C. Hobbs, Asst. Supt. Power Stations, Duquesne Light Co.; and the author.

On motion the meeting adjourned at 10:25 P. M.

K. F. TRESCHOW, *Secretary*.

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, March 5, 1918 at 4:20 P. M. President W. E. Snyder presiding, Messrs. Hawley, James, Neilson, Mott, Pittman, Danforth, Barbour, Duff and the Secretary being present.

The minutes of the last regular meeting held February 5th were read and approved.

The applications of the following gentlemen having been regularly published to the Society, pursuant to the action of the Board, were elected to membership:

MEMBER

Herr, Benjamin M.

ASSOCIATE MEMBERS

Burd, Francis John — Knapp, James Howard

The applications of the following gentlemen were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBERS

Armstrong, Harry Howard — Hiles, John Delvin
Cheeseman, Redick C. — Merz, Paul Caesar

JUNIOR

Johnson, Carl Emil

The Secretary read letters of resignation from the following gentlemen:

Lindstrom, C. A. — Hitchcock, H. K. — McCargo, Grant

After discussion, they were ordered accepted.

The Secretary read a letter of resignation from Mr. A. W. Fischer, Past President of the Society and it was moved and carried that, in view of his connection with the Society as a Past President and a member of long standing, he be exempted from payment of further dues. The Secretary was requested to write him to this effect.

The Secretary announced the death of the following gentlemen:

Bryan, James	Died Feb. 23, 1918
Haines, John L.	Died Dec. 14, 1917
Yardley, Edmund	Died Feb. 18, 1918

The Secretary was requested to ask Mr. J. M. Graves to prepare a memoir for Mr. James Bryan and Mr. F. L. Garlinghouse for Mr. Yardley.

The report of the Secretary showing the financial condition of the Society at the close of business January 31, 1918, having been previously audited by the Finance Committee, was approved.

COMMITTEE REPORTS:

Entertainment

The Entertainment Committee held one meeting during the month and have arranged for an entertainment to be held at the Ft. Pitt Hotel, April 13th, at which time Mr. C. C. Carpenter of Philadelphia will be the principal speaker.

George H. Neilson, *Chairman.*

Finance

Mr. Pittman, Chairman of the Finance Committee reported that the accounts of the Society had been audited Jan. 31st and found in good order.

House

Mr. George H. Barbour, Chairman of the House Committee presented the following report:

The Society Rooms have been kept open during the evenings with a daily attendance of from one to nine, the total for the week varying from 16 to 25, making an average of 20 for the month.

Membership

Mr. H. D. James, Chairman of the Membership Committee presented the following report:

A meeting of the Membership Committee was held on Feb. 28th to discuss the change in the By-Laws, which permit of certain privileges being extended to members of National Societies at the discretion of the Board of Direction. It was the opinion of the Membership Committee that we should recommend to the Board of Direction that they remit the initiation fees of members of the American Society of Mechanical Engineers, American Institute of Electrical Engineers, American Society of Civil Engineers, and the American Institute of Mining Engineers; also that members in these Societies be admitted to a similar grade in the Engineers' Society of Western Pennsylvania.

If these recommendations are approved by the Board, we would ask the privilege of preparing a special application blank to be used by the members of the four National Societies enumerated above.

We discussed the benefits derived from the increase of our permanent fund by payment of initiation fees, but it was our opinion that the most important question at present was to finance the going organization of our Society, which must be done almost entirely by the dues of members. The activities of the Society and its attractiveness to its members will depend largely upon keeping the membership up, and particularly by interesting Engineers who may become active members.

During the war engineers are economizing in order to meet the increasing cost of living. It will, therefore, be necessary to offer special inducements to interest many engineers at the present in joining the Society. We would, therefore, like to have the privilege of remitting the initiation fees to cover the period of the war and until six months after the conclusion of peace. As the present Board has power only to grant this privilege for the present year, it will be necessary to take the matter up again, unless we are so fortunate as to conclude the war at an early date.

H. D. James, *Chairman.*

After discussion, it was moved and carried that the report of the Committee be accepted and the recommendations set forth be put into effect after sufficient time has elapsed to permit appeal as provided in Article 6, Section 2 of the By-Laws.

Civic Affairs Committee

Mr. Samuel E. Duff, Chairman of the Civic Affairs Committee presented the following report:

The Civic Affairs Committee submits the following report:

With the approval of President Snyder, the Committee organized with the following members:

Samuel E. Duff, *Chairman*
K. H. Talbot, *Vice Chairman*
G. M. Lehman
J. H. Minton
G. W. Nichols
Walter Spellmire

It is the intention to add to the working force of the Committee from time to time by creating sub-committees, of which present members of the Committee will be Chairmen. In addition to the Chairman, the sub-committees will consist of members of the Society selected for particular usefulness on the subject to be considered. The sub-committees will work under the direction of the Committee and report to it.

At a meeting held on March 4th, all members of the Committee were present and after a general discussion the following resolution was adopted:

RESOLVED: That the Civic Affairs Committee report to the Board of Direction their unanimous opinion that the relationship of the Engineers' Society of Western Pennsylvania to public affairs properly implies constructive suggestions relative to and impartial criticism of legislation and the expenditure of public funds involving engineering principles and practice. To this end the Committee will maintain a watchful attitude toward legislation and public expenditure with the purpose of bringing to the attention of the Board such matters as appear worthy of consideration.

SAMUEL E. DUFF, *Chairman.*

It was moved and carried that the report as presented be accepted.

Publication

Mr. Danforth, Chairman of the Publication Committee reported that the Committee had held one meeting during the month and had taken up the matter of Mr. McClelland's offer of $45.00 per issue for editing the Proceedings, 10 issues per year. This price also includes the yearly index, which Mr. McClelland has been doing for us for a number of years, for which he usually charged from $10.00 to $15.00. The Committee recommended to the Board of Direction that this offer be accepted, with the understanding that the contract can be terminated on a month's notice.

After discussion, it was moved and carried that the recommendation of the Committee be accepted.

On motion the meeting adjourned at 5:40 P. M.

K. F. TRESCHOW, *Secretary.*

STRUCTURAL SECTION

The regular bi-monthly meeting of the Structural Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg. Tuesday, March 5, 1918 at 8:20 P. M., Chairman George H. Danforth presiding, 20 members and visitors being present.

The minutes of the last regular meeting held January 8th, were read and approved.

The proposed amendments to the By-Laws which had been published to the Section in due form, were submitted for final action, whereupon on motion, they were adopted by a unanimous vote.

No further business coming before the Section, the paper of the evening on "Simplification of Riveted Joint Design" was presented by Mr. H. A. S. Howarth, Associate, Albert Kingsbury, Consulting Engineer, Pittsburgh.

The ensuing discussion was participated in by: Mr. K. H. Talbot, Div. Engr, Promotion Bureau, Universal-Portland Cement Co.; Mr. Edward Godfrey, Struct. Engr, Robert W. Hunt & Co; Mr. H. D. James, Asst. to Mgr. Engineering, Westinghouse Elec. & Mfg. Co; Mr. Albert Kinsbury, Cons. Engr; Mr. Paul S. Whitman, Engr, Riter Conley Co; F. L. Egan, Draftsman, Carnegie Steel Co; Mr. George H. Danforth, Struct. Engr, Jones & Laughlin Steel Co; Mr. George W. Nichols, Engr, S. C. Webb Engineering Co, and the author.

On motion the meeting adjourned at 10:22 P. M.

K. F. TRESCHOW, *Secretary.*

REGULAR MONTHLY MEETING

The 368th regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg, Tuesday, March 26th, at 8:17 P. M., President W. E. Snyder presiding, 64 members and visitors being present.

The Minutes of the last regular monthly meeting held Feb. 19, 1918 were read and approved.

The minutes of the March meeting of the Board of Direction were read, announcing the election of one applicant to the grade of Member and two to the grade of Associate Member; the receipt of five applications for membership.

There being no further business before the Society, the paper of the evening on "Monongahela River Navigation" was presented by Lieut. Col. H. C. Stickle, U. S. Army, Retired, District Engineer, Pittsburgh District.

The ensuing discussion was participated in by: Col. Thomas P. Roberts, U. S. Engineer Office; Mr. W. E. Snyder, Mech. Engr, American Steel & Wire Co; Mr. F. L. Egan, Draftsman, Carnegie Steel Co; Mr. J. L. Kerr, Pilot on Monongahela River, Pittsburgh; Mr. R. A. Cummings, Cons. Engr, Pittsburgh, Pa; Mr. George H. Danforth, Struct. Engr, Jones & Laughlin Steel Co, and the author.

On motion the meeting adjourned at 10:17 P. M.

K. F. TRESCHOW, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, April 2nd, at 4:20 P. M., President W. E. Snyder presiding, Messrs. Hoerr, Neilson, James, Hawley, Pittman, Barbour, Crabtree, Linton, Duff and the Secretary being present.

The minutes of the last regular meeting of the Board of Direction, held March 5th, were read and approved.

The applications of the following gentlemen, having been regularly published to the Society pursuant to the action of the Board, were elected to membership:

MEMBERS

Armstrong, Harry Howard — Cheeseman, Redick C.
Hiles, John Delvin

JUNIOR

Johnson, Carl Emil

The applications of the following gentlemen were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBERS

Rackoff, Adolph A. — Riley, Albert Dowler
Schuttler, Carl Hugo

JUNIOR

Smith, George Arthur

The Secretary announced that Mr. George Adam Paff, who joined the Society September, 1912, and resigned December, 1916, desired to be reinstated, whereupon the Secretary was authorized to replace his name upon the Society rolls.

The Secretary presented letters of resignation from the following gentlemen, which, after discussion, were ordered accepted: L. W. Griffith, Edwin S. Reno, R. A. L. Snyder, O. A. R. Wikander.

The report of the Secretary showing the financial condition of the Society at the close of business February 28, 1918, having been previously audited by the Finance Committee, was approved.

COMMITTEE REPORTS

The Committee on Civic Affairs wishes to report activity in connection with the endorsement of the 1918 Road Program of Allegheny County, and the Recommendations of the Pittsburgh Transit Commissioner, details of which are embodied in reports already considered by you at the Special Meeting March 18th.

No other matters have been taken up since and no meetings of the Committee have been held.

Respectfully submitted,
SAMUEL E. DUFF, *Chairman.*

The Entertainment Committee held two meetings during the past month. Arrangements for the Entertainment on April 13th have been completed. The Committee is having trouble in arranging Inspection Trips and the assistance of the members of the Board is asked in this direction.

Respectfully submitted,
GEORGE H. NEILSON, *Chairman.*

A meeting of the Finance Committee was held on March 30th with two members and the Secretary in attendance. The Secretary's financial statement of February 28th was checked and approved. It will be noted from the report that the excess receipts over expenditures of the current year to February 28th, as compared with the same period last year, shows a net loss of nearly $1,200.00. This is due in part to the remission of dues of members who are in the military service of the Government.

The vote of the Society upon transferring the $586.37 in Building Fund to the Permanent Fund was favorable. This gives us a total of $1,129.71 in the Permanent Fund uninvested. The Finance Committee recommends that $1,000.00 of this amount be invested in Liberty Bonds of the third issue.

Respectfully submitted,
E. W. PITTMAN, *Chairman.*

After discussion, it was moved and carried that the report of the Finance Committee be accepted as read and the Committee be authorized to invest $1,000.00 of the cash now in the Permanent Fund for Liberty Bonds of the third issue.

During the past month the Membership Committee has taken steps to secure a complete list of members of the four national engineering societies in the Pittsburgh district. The Chairman of this Committee visited the Secretaries of the four societies on the 15th ult., and had personal interviews with Mr. Rice, Secretary of the A. S. M. E., and Mr. Hutchinson, Secretary of the A. I. E. E. He saw the Assistant Secretary of the A. I. M. E. and a representative of the Secretary of the A. I. C. E.

He was assured that they considered the members published in the annual list of members as being in good standing in the Society and that it was not their practice to publish any names that their Board of Directors did not consider in good standing.

The various Societies admitted that they were carrying many of their members who had joined the military organization and that there were a

few cases in which members were in arrears for their dues, but have made satisfactory arrangements with the Board for the payment of these dues.

Lists of membership in these various Societies, corrected to January 1st, 1918, are being secured and will be used in the campaign which we are now arranging.

Respectfully submitted,
H. D. JAMES, *Chairman*.

The House Committee beg to submit the following report as to the number of visitors in the Society Rooms in the evenings during the month of March:

First week of March	17
Second week of March	16
Third week of March	15
Fourth week of March	21
Total	69

Respectfully submitted,
GEORGE H. BARBOUR, *Chairman*.

In the absence of the Chairman of the Publication Committee, Mr. Danforth, the Secretary reported that the program for the remainder of the year was filled up and arrangements under way for the program for the 1919 season.

The meeting adjourned at 5:45 P. M.

K. F. TRESCHOW, *Secretary*.

SPECIAL MEETING

A special meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., at 8:20 P. M., President W. E. Snyder presiding, 44 members and visitors being present.

No other business coming before the Society, the paper of the evening, on "County Road Construction and Main Arteries," was presented by Mr. William McClurg Donley, Road Commissioner, Allegheny County.

The ensuing discussion was participated in by: Mr. H. D. James, Asst. to Manager Engineering, Westinghouse Elec. & Mfg. Co.; Mr. C. N. Carten, Asst. Engr., Bureau of Water, City of Pittsburgh; Mr. W. E. Snyder, Mech. Engr., American Steel & Wire Co.; Colonel Thomas P. Roberts, U. S. Engineer Office; Dr. Thomas Turnbull, Jr., Director, Tate, Jones & Co.; and Mr. A. C. Toner, Dist. Engr., Portland Cement Association.

On motion, the meeting adjourned at 10:00 P. M.

K. F. TRESCHOW, *Secretary*.

MECHANICAL SECTION

The regular bi-monthly meeting of the Mechanical Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, April 9th, at 8:25 P. M., Chairman Albert Kingsbury presiding, 12 members and visitors being present.

The minutes of the last regular meeting, held February 5th, were read and approved.

The proposed amendment to the By-Laws of the Section was brought up. After count of voting members present, the Chairman announced that a quorum was not present and, therefore, consideration of the amendment would be postponed until the next meeting of the Section.

No further business coming before the Section, the paper of the evening, on "The Design of Governors, with Special Reference to Small Diesel Engines," was presented by Mr. Arthur B. Lakey, Associate, Albert Kingsbury, Engineer, Pittsburgh, Pa.

The ensuing discussion was participated in by: Mr. F. L. Egan, Draftsman, Carnegie Steel Co.; Mr. George H. Danforth, Struct. Engr., Jones & Laughlin Steel Co.; Mr. L. C. Frohrieb, Secy., Federal Engineering Co.; Mr. Albert Kingsbury, Con. Engr.; and the author.

On motion, the meeting adjourned at 10:20 P. M.

K. F. TRESCHOW, *Secretary*.

REGULAR MONTHLY MEETING

The 369th regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, April 16th, at 8:10 P. M., President W. E. Snyder presiding, 44 members and visitors being present.

The minutes of the last regular meeting held March 26th, and of the special meeting held April 2nd, were read and approved.

The minutes of the April meeting of the Board of Direction were read announcing the election of three applicants to the grade of Member and one to the grade of Junior; also the receipt of three applications for membership.

No further business coming before the Society, the paper of the evening, on "Transit Commission," was presented by Mr. E. K. Morse, Transit Commissioner, City of Pittsburgh.

Written discussion was received from: Mr. Edward Godfrey, Struct. Engr., Robert W. Hunt & Co.

The ensuing discussion was participated in by: Mr. Samuel E. Duff, Con. Engr.; Mr. W. E. Snyder, Mech. Engr., American Steel & Wire Co.; Mr. Louis P. Blum, Blum, Weldin & Co.; Mr. Lee C. Moore, Pres., Lee C. Moore & Co.; Mr. Harry J. Lewis, Con. Engr.; Mr. J. R. Parke, Secy., Allied Boards of Trade and Pittsburgh Commercial Club; Mr. P. N. Jones, V. P. & Gen. Mgr., Pittsburgh Railways Co.; Mr. P. E. Hunter, Pres., Independent Bridge Co., Beaver, Pa.; and the author.

On motion, the meeting adjourned at 10:30 P. M.

K. F. Treschow, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, May 7th, at 4:15 P. M., President W. E. Snyder presiding, Messrs. Hoerr, Neilson, Hawley, Pittman, Duff, Danforth, Stucki, Barbour and the Secretary being present.

The minutes of the last regular meeting of the Board of Direction, held April 2nd, were read and approved.

The applications of the following gentlemen, having been regularly published to the Society pursuant to the action of the Board, were elected to membership:

MEMBERS

Rackoff, Adolph A.
Riley, Albert Dowler
Schuttler, Carl Hugo

JUNIOR

Smith, George Arthur

The applications of the following gentlemen were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBERS

Frew, Harry C.
Gibson, John, Jr.
Lease, Leonard John
Reisinger, Horace W.

JUNIOR

Nuernberg, Arthur

The following applications were received from the local members of the four National Societies:

A.S.C.E.

John Farris, Associate Member
G. R. Johnson, Member
A. E. Prack, Junior
H. H. Robertson, Associate
R. V. Warren, Associate Member

A.I.E.E.

G. A. Benney, Associate
J. L. Crouse, Member
H. K. Hardcastle, Associate
A. A. Morton, Associate
A. B. Reynders, Associate
Edward E. Ross, Associate
R. E. Uptegraff, Associate

A.I.M.E.

R. C. Crawford, Member	W. D. Kirk, Member
C. B. Dutton, Member	H. W. Graham, Junior
Roy H. Davis, Member	R. H. McMillen, Member
W. L. Kann, Associate	John G. Pew, Member
J. J. Forbes, Member	Thomas B. Sturges, Member

A.S.M.E.

S. A. Bockius, Member	G. F. Elliott, Junior
L. B. Breed-Love, Junior	C. B. Mills, Member
W. A. Butler, Member	W. E. Moore, Member
Charles W. Cade, Member	G. F. Murphy, Member
H. W. Townsend, Associate Member	J. Ramsey Speer, Member

It was moved and carried that the above names be published to the Society and that the Secretary notify the Secretaries of the National Societies, giving them a list of the names.

Application for transfer to higher grade was received from J. W. Wilson, who was thereupon transferred to the grade of member.

The Secretary presented letters of resignation from the following gentlemen, which, after discussion, were ordered accepted:

D. S. Beyer	Joined May, 1912—April, 1914
E. J. Billings	Joined Jan., 1912
John N. Reese	Joined Nov., 1912
T. J. Wilkerson	Joined April, 1903

The report of the Secretary showing the financial condition of the Society at the close of business March 31, 1918, having been previously audited by the Finance Committee, was approved.

COMMITTEE REPORTS

Mr. Samuel E. Duff, Chairman of the Civic Affairs Committee, advised that no meeting of the Civic Affairs Committee had been held since the last meeting of the Board and no business was before the Committee.

Mr. George H. Neilson, Chairman of the Entertainment Committee, advised that the Smoker and Entertainment held at the Fort Pitt Hotel on Saturday, April 13th, was attended by 135.

The Committee held one meeting during April. There are several prospects for trips, but nothing definite has been decided upon as yet.

Mr. E. W. Pittman, Chairman of the Finance Committee, reports that he went over the Secretary's financial statement for March and approved same.

The $1000.00 registered Liberty Bond has been purchased as directed.

Mr. Barbour, Chairman of the House Committee, reports that the attendance in the Society Rooms during the month of April was as follows: First week, 19; second week, 14; third week, 16; fourth week, 23. Total, 72.

In the absence of Mr. James, Chairman of the Membership Committee, the Secretary reported that out of about 500 letters mailed to local members of the four National Societies, thirty-three applications had been received to date. The Committee has held no meetings during the month.

Mr. Danforth, Chairman of the Publication Committee, reported that two meetings of the committee had been held during the month, at which matters were taken up in connection with the program for the coming season.

Letter was received from McMillin Printing Company asking for an increase of 15 per cent. above our present contract price for printing the Proceedings and, after discussion, the Committee recommended to the Board of Direction that this increase be granted.

Mr. Danforth stated that, after a thorough investigation of the matter, it was found that the increase was found to be in proportion to the increase in cost of material and labor.

After discussion, it was moved and carried that the report of the Committee be accepted and that they be allowed this increase with the understanding that it was entirely voluntary on the part of the Society and might be terminated at any time. The Secretary was requested to notify McMillin Printing Company to this effect

The Secretary read a letter from the League to Enforce Peace, asking the Society to appoint three delegates to attend the "Win the War for Permanent Peace" convention to be held in Philadelphia May 16-18, under the auspices of the League. It was moved and carried that the President be authorized to appoint these delegates from among certain of our members who will probably be in Philadelphia on this date.

The matter of advertising in the Proceedings was taken up, the suggestion being made that the revenues received from this section of the Proceedings were not as large as they should be. After discussion, it was moved and carried that the President appoint an advertising committee, composed of a chairman and two members, to investigate and devise ways and means by which additional advertising could be secured.

The meeting adjourned at 5:25 P. M.

K. F. Treschow, *Secretary.*

CIVIL SECTION

The regular bi-monthly meeting of the Civil Section of the Engineers' Society of Western Pennsylvania was held in the Applied Industries Bldg., Carnegie Institute of Technology, Tuesday, May 7th, at 8:00 P. M., Chairman George H. Danforth presiding, 103 members and visitors being present.

The minutes of the last meeting, held March 5th, were read and approved.

No further business coming before the Section, Prof. W. Trinks, Professor, Mechanical Engineering, Carnegie Institute of Technology, introduced the subject for the evening—"The Training of Mechanics for Maintenance and Repair of Airplanes and Airplane Engines."

Short addresses were made by: Prof. J. C. Sproule, Assoc. Professor, Mechanical Engineering, Carnegie Institute of Technology; Prof. A. H. Blaisdell, Instructor, Department of Mechanical Engineering, Carnegie Institute of Technology; Lieut. W. A. Hammond, Chief Instructor, Airplanes and Airplane Engines, Carnegie Institute of Technology.

Mr. Duff made a motion that a vote of thanks be extended to the gentlemen who so kindly addressed the Society on this subject.

After adjournment, members were taken through the various shops where the different branches of the work which had been described were carried on.

K. F. TRESCHOW, *Secretary.*

REGULAR MONTHLY MEETING

The 370th regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade bldg., May 21st, 1918, at 8:20 P. M., President W. E. Snyder presiding, 47 members and visitors being present.

The minutes of the last regular meeting, held April 16th, were read and approved.

The minutes of the May meeting of the Board of Direction were read announcing the election of three applicants to the grade of Member and one to the grade of Junior and the receipt of five applications for membership. Also the receipt of thirty-two applications for membership in response to letters of invitation to the local members of the four National Societies to join the Society.

No further business coming before the Society, the paper of the evening, on "The Caisson Method for Foundations and Mine Shafts," was presented by Mr. George R. Johnson, District Manager, The Foundation Co., Pittsburgh, Pa.

The ensuing discussion was participated in by: Mr. W. A. Weldin, Blum, Weldin & Co., Pittsburgh, Pa.; Mr. C. T. Day, Draftsman, American Bridge Co., Pittsburgh, Pa.; Mr. W. E. Snyder, Mech. Engr., American Steel & Wire Co., Pittsburgh, Pa.; Mr. George H. Danforth, Struct. Engr., Jones & Laughlin Steel Co., Pittsburgh, Pa.; and the author.

On motion, the meeting adjourned at 9:55 P. M.

K. F. Treschow, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, June 4, 1918, at 4:25 P. M., Vice President George H. Neilson presiding, Messrs. James, Diehl, Danforth, Pittman, Linton, Barbour, Hawley, Hoerr and the Secretary being present.

The minutes of the last regular meeting of the Board of Direction held May 7, 1918, were read and approved.

The applications of the following gentlemen having been regularly published to the Society, pursuant to the action of the Board, were elected to membership:

MEMBERS

Bockius, S. A. (A. S. M. E.)
Butler, W. A. (A. S. M. E.)
Cade, Chas. W. (A. S. M. E.)
Crouse, J. L. (A. I. E. E.)
Crawford, R. C. (A. I. M. E.)
Davis, Roy H. (A. I. M. E.)
Dutton, C. B. (A. I. M. E.)
Forbes, J. J. (A. I. M. E.)
Frew, Harry C.
Gibson, John Jr.
Johnson, G. R. (A. S. M. E.)
Kirk, W. D. (A. I. M. E.)
Lease, Leonard John
McMillen, R. H. (A. I. M. E.)
Mills, C. B. (A. S. M. E.)
Moore, W. E. (A. S M. E.)
Murphy, C. F. (A. S. M. E.)
Pew, John G. (A. I. M. E.)
Reisinger, Horace W.
Speer, J. Ramsey (A. S. M. E.)
Sturges, Thomas B. (A. I. M. E.)

ASSOCIATE MEMBERS

Farris, John (A. S. C. E.)
Townsend, H. W. (A. S. M. E.)
Warren, R. V. (A. S. C. E.)

ASSOCIATES

Benney, G. A. (A. I. E. E.)
Hardcastle, H. K. (A. I. E. E.)
Kann, W. L. (A. I. M. E.)
Morton, A. A. (A. I. E. E.)
Reynders, A. B. (A. I. E. E.)
Robertson, H. H. (A. S. C. E.)
Ross, Edward E. (A. I. E. E.)
Uptegraff, R. E. (A. I. E. E.)

JUNIORS

Breed Love, L. B. (A. S. M. E.)
Elliott, G F. (A. S. M. E.)
Graham, H. W. (A. I. M. E.)
Nuernberg, Arthur
Prack, A. E. (A. S. C. E.)

The following applications were received and their names ordered published to the Society. Of this number, one was received from the A. S. M. E., two from the A. S. C. E. and one from the A. I. E. E. local members:

MEMBERS

Bole, Harry Alan	Carroll, J. G. (A. I. E. E.)
Borgman, William I.	Elliott, James R. (A. S. C. E.)
Brooks, Joseph Bradford	Feucht, George C.
Chester, Walter Durst	Stenersen, Stener

ASSOCIATE MEMBERS

Geddis, Robert H. (A. S. M. E.)	Todd, C. L. (A. S. C. E.)

Application for transfer to higher grade was received from Earl Carlton Sleeman, who was thereupon transferred to the grade of Member.

A letter of resignation was received from Mr. R. C. Warner, which after discussion, was ordered accepted.

The report of the Secretary showing the financial condition of the Society at the close of business April 30, 1918, having been previously audited by the Finance Committee, was approved.

COMMITTEE REPORTS

Mr. Neilson, Chairman of the Entertainment Committee reported that no meetings had been held during the past month, but that the question of an excursion on the river, June 21st, was under consideration. He also stated that the Committee had gone into the matter thoroughly and found that the cost would amount to approximately $350.00. It is probable that the sale of tickets would not cover this amount and the matter was left to the Board to decide as to whether or not the excursion should be held.

After discussion, it was moved and carried that the Committee be authorized to complete arrangements for the excursion on the above date. It was suggested that $1.75 be charged for tickets.

Mr. Pittman, Chairman of the Finance Committee, reported as follows: The Secretary's financial statement of April 30th was examined and found correct. Receipts for the last three months have been normal.

It is recommended that the Society's books be audited by certified accountants this year as was done last year. The cost of this service is about $100.00.

After discussion, it was moved and carried that the report be accepted and the Chairman authorized to have the books audited as of July 1, 1918, cost to be left entirely to him.

Mr. Barbour, Chairman of the House Committee, reported the following attendance in the Society Rooms in the evenings for the month of May.

First week	13
Second week	14
Third week	20
Fourth week	12
	59

Mr. James, Chairman of the Membership Committee, reported that no meetings had been held since the last Board meeting, but several of the Committee had been following up the members of the National Societies endeavoring to get them to accept our invitations to join. Four additional acceptances received since the last Board meeting, two from the A. S. C. E., one from the A. I. E. E. and one from the A. S. M. E.

Mr Danforth, Chairman of the Publication Committee, reported that a tentative program had been arranged for the coming year.

The Secretary read a letter from Mr. C. E. Skinner enclosing copy for a resolution adopted by the Engineering Council in regard to a clause of the Army and Navy Appropriation Bills prohibiting the payment to public employees or to employees of private establishments under Government control any cash reward premium or bonus for superior service, protesting against the adoption of such a measure.

It was moved and carried that the Secretary write Mr. Skinner thanking him for his interest and state that the Board agreed with the above mentioned resolution.

The Secretary read a letter from the Secretary of the Detroit Section of the American Society of Mechanical Engineers, enclosing copy of a resolution adopted by that Section, also the Detroit Engineering Society in joint session, May 3, 1918, requesting that universities, colleges and technical schools throughout the country be asked to provide special courses of instruction for women students in drafting, tracing, inspection and testing of materials, both physically and chemically, in order to meet the shortages in skilled workers in the trades and professions and to release men for other duties in connection with the war.

After discussion, it was moved and carried that the President be authorized to appoint a committee of three to look into this matter and report at the next regular meeting of the Board of Direction any action the Society might take to promote this work among the schools in this district.

On motion the meeting adjourned at 5:20 P. M.

K. F Treschow, *Secretary.*

SPECIAL MEETING

A special meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Building, Tuesday, June 4, 1918, at 8:28 P. M., Chairman E. W. Pittman presiding, 28 members and visitors being present.

No business coming before the Society, the paper of the evening on "The Promotion of Industrial Research" was presented by Dr. John Johnston, Secretary, National Research Council, Washington, D. C.

The ensuing discussion was participated in by: Prof. Horace R. Thayer, Assoc. Professor, Structural Design, Carnegie Institute of Technology; Mr. Paul S. Whitman, Engr., Riter Conley Co., Pittsburgh, Pa.; Mr. C. E. Skinner, Research Division, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa., and the author.

It was moved, seconded and carried unanimously that a vote of thanks be extended to Dr. Johnston for his very interesting paper on this subject.

On motion the meeting adjourned at 9:35 P. M.

K. F Treschow, *Secretary.*

PERSONAL ITEMS

William T. Manning at one time Chief Engineer of the Baltimore & Ohio Railroad Co. and well known in Pittsburgh, died suddenly at Baltimore, Md., July 8th last.

Mr. Manning was well known among the early railroad engineers in this vicinity, having participated in the construction of the Chicago Division of the Baltimore & Ohio as well as the Somerset & Cambria Branch of that road.

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, Sept. 3, 1918, at 4:15 P. M., President W. E. Snyder presiding, Messrs. Hoerr, Neilson, Duff, Stucki, Pittman, Danforth, Diehl, Hawley, James and the Secretary being present.

The Minutes of the last regular meeting held June 4th were read and approved.

The applications of the following gentlemen having been regularly published to the Society, pursuant to the action of the Board, were elected to membership:

MEMBERS

Bole, Harry Alan
Borgman, William
Brooks, Joseph Bradford
Carroll, J. G. (A. I. E. E.)
Chester, W. D.
Elliott, James R. (A. S. C. E.)
Feucht, George C.
Stenersen, Stener

ASSOCIATE MEMBERS

Geddis, Robert H. (A. S. M. E.)
Todd, C. L. (A. S. C. E.)

The following applications were received and their names ordered published to the Society. Of this number, three were received from the A. S. M. E. local members and one from the A. S. C. E.

Grant, Charles A.....................Associate member A. S. M. E.
Haynes, A. W., Jr...................Associate member A. S. M. E.
Moore, H. Lee..Junior A. S. M. E.
Palmer, Charles Skeele
Schultz, A. L...Member A. S. C. E.
Sprager, Samuel Olter
Trax, Edward Carey

Applications for transfer to higher grade were received from the following gentlemen, which after discussion, were transferred as designated:

Barnes, Hugh Cooper..Member
Frank, William Klee..Member
Hill, G. May...Member
Noble, Robert Elliott..Member
Tayman, George S......................................Associate Member

Letters of resignation were received from the following gentlemen, which after discussion, were ordered accepted:

Neale, Richmond H.	Joined Dec. 1916
Reeser, E. B.	Joined Dec. 1915
Rice, George S.	Joined Apr. 1913
Schwartz, Benjamin	Joined Sept. 1916
Sloane, D. M.	Joined Dec. 1909
Sprague, N. S.	Joined Apr. 1908

The Secretary announced the death of the following gentlemen:

Obermeyer, W. S.	
Kidd, W. S.	Died June 20, 1918
Langley, John W.	Died May 10, 1918

It was moved and carried that the President appoint a Committee of three to prepare a memoir of Dr. Langley, who was an honorary member of the Society since 1893.

The reports of the Secretary showing the financial condition of the Society at the close of business May 31st and June 30th, 1918, having been previously audited by the Finance Committee, were approved.

COMMITTEE REPORTS

Mr. Duff, Chairman of the Civic Affairs Committee, reported that the Committee had held no meetings during July and August, therefore had nothing to report.

Mr. Neilson, Chairman of the Entertainment Committee, reported that the Committee had held no meetings during July or August, and had nothing to report.

Mr. Pittman, Chairman of the Finance Committee, reported that the books of the Society had been audited by Ernst & Ernst for the period of July 1st, 1917, to July 1st, 1918, in accordance with the Board's authorization and were found to be in good condition.

Mr. Barbour, Chairman of the House Committee, reported an evening attendance of 65 during the months of June, July and August.

Mr. James, Chairman of the Membership Committee, reported that no meetings of the Committee had been held during the Summer months.

Mr. Danforth, Chairman of the Publication Committee, reported verbally, stating that no meetings had been held during the Summer months, but that arrangements were being made for the program for the coming season.

The committee has under consideration the matter in regard to stenographer for the Society meetings and requests that they be given another month before making a final report.

Mr. Mott, Chairman of the Special Committee on Training of Women for Technical Positions, reported as follows:

As Chairman of your Committee appointed to report on the question of training women for technical positions, I beg to report the progress.

Mr. John A Hunter and Dr. Francis Tyson are the other members of the Committee but it has been impossible for us to hold a joint meeting because of absence from town of one or another member of the Committee. I beg, therefore, to suggest that the Commitee be continued for another month and I trust that we shall be able to present a report at the October meeting.

One point, in particular, has come to our attention which I may mention at this time. It seems desirable that the members of the Society should make known to your Committee, through the Secretary, just what positions of a technical nature they have in their organizations where the service of women would be effective.

Respectfully submitted,

(Signed) Wm. E. Mott, *Chairman.*

In connection with the last paragraph of this report, it was moved and carried that a notice be put on the next announcement, explaining the object of the appointment of the committee and asking all members of the Society to advise the Secretary what technical positions in their respective companies women should be trained to hold.

Mr. A. Stucki, Chairman of the Special Committee on Advertising, in the Proceedings, reported that he had been in communication with several men in regard to their securing advertising and stated he would go into the matter thoroughly during the coming Fall and hoped to have something definite to report to the Board in the near future.

MISCELLANEOUS

The Secretary presented a letter from the American Society of Mechanical Engineers in regard to organization of By-Laws of Local Sections. The letter was ordered filed.

It was suggested that the name and seal of the Society be placed on some of the windows of the Club Rooms and the matter was referred to the House Committee with power to act.

The Secretary asked permission of the Board to have a man from Ernst & Ernst, who have just made an audit of the books, to assist him a day or two in making several suggested changes in keeping the Society accounts. It was moved and carried that he be authorized to secure this assistance.

The meeting adjourned at 5:30 P. M.

K. F. Treschow, *Secrtary.*

SPECIAL MEETING

A special meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, Sept. 17, 1918, at 7:30 P. M., President W. E. Snyder presiding, Messrs. Hoerr, Duff, James, Stucki, Pittman, Barbour and the Secretay being present.

President Snyder stated that the meeting had been called to place before the Board for discussion a letter received from Mr. C. E. Drayer, Secretary of the American Association of Engineers in regard to joint membership between the American Association of Engineers and the Engineers' Society of Western Pennsylvania and suggesting that Mr. W. H. Finley, President of the Association be invited to address our Society on the plan of joint membership.

After considerable discussion, it was moved and carried that the Secretary be requested to write Mr. Drayer advising him that the Board did not feel, in view of war conditions in this district, that this was an opportune time to increase the activities of the Society.

Mr. Samuel E. Duff, Chairman of the Civic Affairs Committee, presented the following report in regard to an appropriation of $40,000 made by City Council and County Commissioners for a housing survey in the Pittsburgh District.

To the Board of Direction,

Engineers' Society of Western Penna.:

Dear Sirs:

Your Committee on Civic Affairs respectfully calls to your attention the appropriation of $40,000 by the City Council and the County Commissioners for a housing survey of the Pittsburgh District.

The survey is to be made to comply with the request from the United States Department of Labor for information as to housing conditions in this district, as affected by the construction of the Ordnance Plant on Neville Island and other increased facilities for the production of war necessities.

Your Committee feels that the Engineers' Society should heartily endorse this project, and render every possible aid to the authorities so that accurate and complete data shall be promptly obtained from which measures effective for the present emergency as well as for the permanent improvement of the community may be adopted and carried out.

Transportation is the most important single factor in the housing problem. Your Board has already examined and approved the Report of the Transit Commissioner of Pittsburgh, and urged the City authorities to promptly carry out the Commissioner's recommendations for street widening and automobile parking.

The Civic Affairs Committee, therefore, respectfully suggests that the Board address letters to the Mayor and Council of the City of Pittsburgh and to Commissioners of Allegheny County pointing out the advisability of using as a basis for a study of the housing problem the very complete and accurate data obtained by the Transit Commissioner as to the place of residence of the employees of large corporations, extended by additional studies to be made by the Commissioner to cover new conditions.

Your Committee also suggests that the Board call the attention of the Civil authorities to the evident saving in time and expense certain to be obtained if the housing survey is made under the direction of the Transit Commissioner and by engineers familiar by residence in this community with the unusual problems involved in housing and transportation arising from the topographic and meteorological conditions of the territory involved,

Respectfully submitted,

Civic Affairs Committee,
Samuel E. Duff, *Chairman;*
J. H. Minton,
George W. Nichols,
W. B. Spellmire.

After discussion, it was moved and carried that the report be approved and that action be taken as stated therein.

The meeting adjourned at 8:00 P. M.

K. F. Treschow, *Secretary.*

REGULAR MONTHLY MEETING

The 371st regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, Sept. 17, 1918, at 8:15 P. M., President W. E. Snyder presiding, 34 members and visitors being present.

1. The Minutes of the last regular meeting held May 21 and of the special meeting held June 4th were read and approved.

2. The Board of Direction reported the election of ten applicants to membership, eight to the grade of Member and two to that of Associate Member, and the receipt of seven applications for membership.

3. Announcment was made of the following appointments to the Nominating Committee, who are to nominate officers for 1919.

Sydney Dillion, *Chairman;*
W. L. Affelder,
Frederic Crabtree,
Wm. F. Hall,
J. A. McEwen.

4. There being no further business, the paper of the evening on "Heating Liquids by Electricity" was presented by Mr. H. O. Swoboda, Consulting Engineer, Pittsburgh, Pa.

Ensuing discussion was presented by Mr. Samuel E. Duff, Cons. Engr., Pittsburgh, Pa.; Mr. W. E. Snyder, Mech. Engr., American Steel & Wire Co., Pittsburgh, Pa., and the author.

On motion the meeting adjourned at 9:45 P. M.

K. F. Treschow, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania, was held in the Society Rooms, Union Arcade Building., Tuesday, October 1, 1918, at 4:20 P. M., President Snyder presiding, Messrs. Hoerr, Duff, Neilson, Pittman, Danforth, Crabtree, Mott, Linton, Hawley, James and the Secretary being present.

The Minutes of the last regular meeting held September 3, and of the Special Meeting held September 17, were read and approved.

The applications of the following gentlemen having been regularly published to the Society, pursuant to the action of the Board were elected to membership.

MEMBERS

Palmer, Charles Skeele — Schultz, A. L. (A. S. C. E.)

ASSOCIATE MEMBERS

Trax, Edward Carey

Grant, A. W., Jr. (A. S. M. E.) — Haynes, Charles H. (A. S. M. E.)

JUNIORS

Moore, H. Lee (A. S. M. E.) — Sprager, Samuel Olter

The following applications were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBERS

Blake, Alfred E. — Farmer, Homer Gilbert

Van der Pyl, Edward

JUNIORS

Cooley, Herbert M. — Sildat, John George

Applications for transfer to higher grade of membership was received from John R. Heath, who after discussion was transferred to the grade of Member.

Letters of resignation were received from the following gentlemen, which after discussion were ordered accepted:

Wheeler, Charles V.......................................Joined Nov., 1900
Fogg, L. W...Joined Mar., 1900

Letter of resignation presented by Mr. G. O. Loeffler. Prof. Crabtree stated that he knew Mr. Loeffler personally and would see him within the next few days in regard to withdrawing his resignation.

Letter of resignation was presented from Mr. Emil Swensson who has been a member of the Society since 1887. After discussion, it was moved and carried that Mr. Swensson's dues be remitted until further notice and that the President appoint a Committee to write a letter to Mr. Swensson advising him, that in view of his long membership in the Society and his many years of hard work on its behalf, the Board did not feel that the Society could afford to have his name removed from the Society rolls.

COMMITTEE REPORTS

Mr. Duff, Chairman of the Civic Affairs Committee, reported that a meeting had been held September 16 to consider the appropriation for a housing survey made by the City of Pittsburgh and Allegheny County. On September 17 a meeting was held and report made to the Board of Direction concerning this matter. No other meetings were held during the meeting and no business is now before the Committee.

Mr. Neilson, Chairman of the Entertainment Committee, reported that the Committee is working on a "Get-Together" entertainment to be held the latter part of October. Also, that the Committee had several prospects for inspection trips which would be announced later.

Mr. Pittman, Chairman of the Finance Committee, reported that no meeting of the Finance Committee had been held. The Secretary's statements for July and August had been checked and approved by the chairman.

Mr. Barbour, Chairman of the House Committee, reported that the evening attendance for September was 19.

Mr. James, Chairman of the Membership Committee, reported for the committee as follows:

A meeting of the Membership Committee was held at the Society Rooms on Wednesday, Sept. 25th. The procedure for admitting members of the four national societies, approved by the Board, was discussed. The following items were agreed upon by the Membership Committee, on which they wish the Board's approval:

Our request for the omission of the initiation fee when admitting these members was based upon the member being admitted to the corresponding grade in the Engineers' Society of Western Pennsylvania. A check at that time indicated that this recommendation was perfectly safe, as the requirements in the national societies were at least equivalent to the requirements in our Society for corresponding grades. The Committee wishes to further recommend that when an engineer is admitted to our Society, and in the opinion of the Board of Direction is entitled to a grade higher than his equivalent in the national society, the Secretary request the member to apply for transfer to this higher grade at the time he is notified of his election to our Society.

The question of what constitutes a member in good standing of a national society has been investigated by your Committee. Your Chairman had a personal interview with the Secretary's office of each of the four

national societies. The same statement was made by each office, namely, that they only published in their annual list the names of members who were in good standing. At the same time we furnished them a list of members to check which showed the same results. The Membership Committee, therefore, recommends that the latest annual list be used for checking a man's standing in his society and that we discontinue writing to the Secretaries of the national societies for each member elected.

Some of the chemical engineers in Pittsburgh have requested that they also be admitted to our Society without initiation fee. The American Chemical Society is the largest of their national organizations. It has about 12 000 members. The Electro Chemical Society has about 2000 members, and the Chemical Engineers about 1000 members. It is, therefore, evident that if this privilege is extended to the chemical engineers the American Chemical Society should be the national society which we recognize. This society has only one active grade of membership. It also has Juniors, which correspond to our Student Juniors.

The Membership Committee recommend that members of the American Chemical Society be admitted as associates of the Engineers' Society of Western Pennsylvania without the payment of an initiation fee, provided they are in good standing in their society at the time of election. Where the chemist is known to have qualifications for a higher grade of membership in our Society, and the Board indicates the higher grade to which he is entitled, the Secretary should request him to make application for transfer to the higher grade at the time he is notified of his election as an associate.

It was moxed and carried that the report of the Membership Committee be approved and action taken as stated therein; also that the Committee be appointed to look into the matter of organizing a Chemical Section of the Society.

Mr. Danforth, Chairman of the Publication Committee, reported as follows:

The Publication Committee has held no meetings during the past month. There has been a number of difficulties developed in connection with our program, due to the fact that it seems to be impossible to get anybody definitely committed to a formal paper under existing conditions, and the requests for delays are very numerous. On this account it will be necessary to invert the order of section papers and have the Civil Section in October in place of the Mechanical Section, the Mechanical Section replacing the Civil Section in November.

Prof. Mott, Chairman of the Special Committee on Training of Women for Technical Positions reported progress.

The question of remission of entrance fees for local members of certain societies was taken up and discussed and it was moved and carried, with one negative vote, that the Membership Committee be requested to consider including in this list the Society for Promotion of Engineering Education.

The meeting adjourned at 5:45 P. M.

K. F. Treschow, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, November 12, 1918, at 4:15 P. M., President W. E. Snyder presiding, Messrs. Duff, Danforth, Pittman, James, Crabtree, Hawley, Hoerr, Stucki and the Secretary being present.

The Minutes of the last meeting held October 1st were read and approved.

The applications of the following gentlemen having been regularly published to the Society, pursuant to the action of the Board, were elected to membership:

MEMBERS

Blake, Alfred E. Farmer, Homer Gilbert
Van der Pyl, Edward

JUNIORS

Cooley, Herbert M. Sildat, John George

The following applications were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBERS

Landell, John Adams Brunro Sadler, Jr., Orin Winslow
Philipsen, Thorulf

JUNIORS

Reed, Charles M.

Letters of resignation were presented by the following gentlemen and after discussion, they were ordered accepted:

S. M. Marshall........................Joined February, 1912
G. C. Saulters..........................Joined October, 1913
W. J. R. Rector....................Joined February, 1913

The Secretary reported the death of Mr. William Kent who was the first treasurer of the Society. Mr. James H. Harlow the first secretary.

Mr. Hawley agreed to write a memoir of Mr. James H. Harlow and Mr. Snyder one of Mr. Kent.

The report of the Secretary showing the financial condition of the Society at the close of business September 30th, having been previously audited by the Finance Committee, was approved.

In the absence of Mr. Sydney Dillon, Chairman of the Nominating Committee, Prof. Crabtree presented the following report:

Pittsburgh, Pa., Nov. 2, 1918.

To the Board of Direction
Engineers' Society of Western Penna.

Gentlemen:

At a meeting of the Nominating Committee of the Engineers' Society of Western Pennsylvania, held in the Society Rooms, Saturday afternoon, November 2nd, the following gentlemen were unanimously chosen for the respective offices indicated below:

For President.............George H. Neilson
For Vice President.............George H. Danforth
For Treasurer.............A. Stucki
For Directors............. W. B. Spellmire, F. C. Schatz

Respectfully submitted,

Sydney Dillon, Chairman
J. A. McEwen
Frederic Crabtree
Wm. F. Hall
W. L. Affelder

It was moved and carried that the nominations be approved and the names published to the Society in accordance with the By-Laws.

COMMITTEE REPORTS

Mr. Duff, Chairman of the Civic Affairs Committee, reported that the Committee on Civic Affairs had held no meeting since their last report and that there was no business before the committee.

Mr. Neilson, Chairman of the Entertainment Committee, reported that no meetings had been held during October, but held one meeting November 9th, to make arrangements for a Smoker, which will be held November 22nd, at the Ft. Pitt Hotel.

Mr. Pittman, Chairman of the Finance Committee, reported as follows:

As authorized by the Board, Ernst & Ernst, accountants, have been called into consultation with a view of improving our methods of accounting. In cooperation with the Secretary they have wrought important changes in our methods and simplified our records. All essential records that were formerly in three books are now consolidated in one book, well adapted to the purpose. The old form of checks have now been superseded by voucher checks. These changes will facilitate the work of the Secretary and the finance committee, and will lighten the labor of an annual audit of our books.

The Secretary has sent out personal letters in an effort to collect back dues from a number of members, and it is hoped that these letters will increase our revenue.

We are still carrying on our membership list the names of between 30 and 40 men who have not paid dues for the years 1914, 15, 16, 17 and 18. It is recommended by the Finance Committee that these men be advised that their names will be dropped from our rolls if their back dues remain unpaid after January 1, 1919.

There are eleven men who have made application for membership, accompanied by a preliminary payment of $5.00, who have not made any subsequent payment. It is recommended that these men also be dropped from our records.

In regard to the third and fourth paragraphs of the above report, it was moved and carried that all members who owe for the years 1914, 15, 16, 17 and 18, with the exception of a few names mentioned, be notified by the Secretary that unless their dues are paid by January 1, 1919, they will be dropped from the rolls of the Society in accordance with the By-Laws.

In regard to the eleven men who failed to matriculate, it was suggested that the Secretary go over this list and call the endorsers in order to ascertain whether or not the men are still in this district; those which we are unable to locate to be dropped from the Society Rolls, Jan. 1, 1919, unless their dues are paid previous to that time.

Mr. Barbour, Chairman of the House Committee, reported the following evening attendance during October:

First week	10
Second week	20
Third week	7
Fourth week	12
Last four days	10
	59

Mr. Danforth, Chairman of the Publication Committee, reported verbally, stating that no meetings of the Committee had been held on account of the epidemic. The Committee was obliged to cancel all meetings to date.

The program for some time to come is in very good shape and a number of papers are being kept in reserve in case any of the scheduled authors find it impossible to present their papers on the date set.

At the request of the President, the Secretary read the following letter received from Mr. A. H. Krom, Director of Engineering, U. S. Department of Labor:

To the President,
Engineers' Society of Western Penna.,
Pittsburgh, Pa.

Dear Sir:

In view of the great importance of the technical man in the present war and the necessity of conserving his powers, I hereby urge the officers of all technical societies to exert their official influence to continue the activities which they represent. Don't be discouraged by the fact that your membership is being decreased by enlistments and war demands. Strengthen your organization and keep your members acquainted with matters relating to their welfare and advancement. Originate plans for professional development.

Be ready with society service when our heroes return. Now, as never before, the engineer and the engineering profession are conceded that place in the world's estimation so long coveted and deserved. Carry this war-time recognition into the pursuits of peace by shouldering your share of civic responsibility and make the Society, headquarters for such action.

I would also be glad to have you present this matter properly and as soon as possible at a meeting of your Society and to advise me in regard to any action you take.

Yours very truly,
(Signed) A. H. Krom,
Director of Engineering.

It was suggested that this letter be read before the Civil Section this evening and the next two or three meetings of the Society.

The meeting adjourned at 5:30 P. M.

K. F. Treschow, *Secretary.*

SPECIAL MEETING

A special meeting of the Board of Direction of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, November 19, 1918, at 10:00 P. M., President W. E. Snyder presiding, Messrs. Duff, Pittman, Stucki, Hoerr, James, Hawley, Danforth and the Secretary being present.

The President stated that he had called the meeting for the purpose of taking action on the Resolution presented by the Westinghouse Memorial Committee and approved at the regular monthly meeting of the Society held Tuesday, November 19th.

After discussion the following resolution was presented by Mr. Samuel E. Duff, seconded by Mr. A. L. Hoerr and carried unanimously:

Whereas, At the regular monthly meeting of the Society held this evening, the following resolution was adopted:

"Resolved, That the report of progress of the Westinghouse Memorial Committee, dated November 19, 1918, be received and filed; that the Committee be continued; and that the proper officers of the Engineers' Society of Western Pennsylvania be hereby authorized and instructed to acquire the Westinghouse Property and to convey the same to the City of Pittsburgh for a public park and Memorial to the late George Westinghouse, without financially involving the Society, as recommended by the Committee."

RESOLVED, That pursuant to the above resolution of the Society, the Board of Direction does hereby authorize and direct the proper officers of the Society to execute and deliver a deed conveying to the City of Pittsburgh the property of the late George Westinghouse for a public park upon the terms and conditions suggested by the Westinghouse Memorial Committee and approved by the Society, at its meeting held this day, and the Board of Direction does hereby constitute and appoint Kenneth F. Treschow to be its attorney, for it and in its name and as and for its corporate act and deed to acknowledge this deed before any person having authority by the laws of the Commonwealth of Pennsylvania to take such acknowledgement to the end that it may be duly recorded.

On motion the meeting adjourned at 10:30 P. M.

K. F. TRESCHOW, *Secretary.*

CIVIL SECTION

The regular bi-monthly meeting of the Civil Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, November 12th, 1918, at 8:12 P. M. Chairman J. A. McEwen presiding, in the absence of the Chairman and Secretary who were attending a special meeting of the Board, 43 members annd visitors being present.

The Minutes of the last meeting held May 7, 1918, were read and approved.

The following letter was read from the Director of Engineering, Dept. of Labor, United States Employment Service:

To the President,
Engineers' Society of Western Penna.,
Pittsburgh, Pa.

Dear Sir:

In view of the great importance of the technical man in the present war and the necessity of conserving his power, I hereby urge the officers of all technical societies to exert their influence to continue the activities which they represent. Don't be discouraged by the fact that your membership is being decreased by enlistements and war demands. Strengthen your organization and keep your members acquainted with matter relating to their welfare and advancement. Originate plans for professional development.

Be ready with Society service when our heroes return. Now, as never before, the engineer and the engineering profession are conceded that place in the world's estimation so long coveted and deserved. Carry this war-time recognition into the pursuits of peace by shouldering your share of civic responsibilities and make the Society, headquarters for such action. I would be glad to have you present this matter properly and as soon as possible at a meeting of your Society and to advise me in regard to any action you take.

Yours very truly,
(Signed) A. H. KROM,
Director of Engineering.

No further business coming before the Section, Mr. James S. Martin, Engineer, Pittsburgh Railways Co., presented a paper on "New and Little Known Methods of Calculation of Beams, Girders and Arches." The ensuing discussion was participated in by: J. A. McEwen, Sales Mgr., Pittsburgh Bridge & Iron Works, Pittsburgh, Pa.; Prof. H. R. Thayer, Assoc. Professor., Structural Design, Carnegie Institute of Technology; George W. Nichols, Engr., S. C. Webb Engineering Co., Pittsburgh, Pa., and the author.

On motion the meeting adjourned at 10:25 P. M.

K. F. TRESCHOW, *Secretary.*

REGULAR MONTHLY MEETING

The 372nd regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldge., Tuesday, November 19, 1918, at 8:10 P. M., President W. E. Snyder presiding, 82 members and visitors being present.

The Minutes of the last regular meeting held September 17th were read and approved.

The Board of Direction reported the election of three applicants to the grade of Member and two to the grade of Junior and the receipt of four applications for membership.

Under the head of new business, Mr. William L. Scaife, a member of the Special Committee on the Westinghouse Memorial, in the absence of Mr. George S. Davison, Chairman, presented the following report together with the accompanying resolution:

At the regular monthly meeting of the Engineers' Society of Western Pennsylvania held October 25, 1916, a Committee was appointed to be known as the WESTINGHOUSE MEMORIAL COMMITTEE. Its duties were to investigate a plan for the purchase of the former home of Mr. George Westinghouse, and for erecting on the property a suitable memorial,—the whole to be turned over to the City of Pittsburgh under the guarantee that this gift should receive the same care and protection as other parks owned by the city.

The Committe which was appointed comprised Messrs. George S. Davison, Chairman; Julian Kennedy, William L. Scaife, Charles F. Scott and E. B. Taylor. This Committee now desires to make a progress report.

The work of the Committee has proceeded so far that the financial arrangements necessary for the purchase of the land have been completed and an informal conference has been held with the City Council to discuss the methods and conditions under which the property involved would pass into the possession of the City of Pittsburgh.

Among the subscribers to the funds necessary for the purchase of this property are the Westinghouse Air Brake Company and the Westinghouse Electric & Manufacturing Company. It has been suggested by representatives of these companies that the title to the property (as it passes from Mr. George Westinghouse, Jr., to the City of Pittsburgh) should pass through the Engineers' Society of Western Pennsylvania. This would involve Mr. Westinghouse's making a deed to this Society in which the full compensation for the property would be mentioned but in the payment of which the Society would not in any way be involved. The title

would then pass through the Society, by action of its officers, to the City of Pittsburgh, the consideration in this case being such restrictions as would go with the property to secure the care and protection for the property suggested in the original resolution providing for the appointment of this Committee.

The City Attorney is now at work framing an ordinance covering the matter. This ordinance will be introduced into the City Council at its regular meeting, November 25th. If the ordinance be acceptable at that time, it is believed that final action will be taken not later than December 3rd of this year.

The question of a suitable memorial to be erected on this property will be a matter for further consideration.

This Committee requests the Society to take such action at this meeting as will give the officers of the Society authority to acquire the property and convey it to the City of Pittsburgh.

Respectfully submitted,
George S. Davison, Chairman,
Julian Kennedy
William L. Scaife
Charles F. Scott
E. B. Taylfr

Resolved, That the report of the progress of the Westinghouse Memorial Committee dated November 19, 1918, be received and filed; that the Committee be continued; and that the proper officers of the Engineers' Socity of Western Pennsylvania be hereby authorized and instructed to acquire the Westinghouse property, and to convey the same to the City of Pittsburgh for a public park and Memorial to the late George Westinghouse, without financially involving the Society, as recommended by the Committee.

It was moved by Mr. William L. Scaife and seconded by Mr. Samuel E. Duff and carried unanimously, that the resolution be adopted and the report of the Committee be accepted as read.

No further business coming before the Society ,the paper of the evening on "The Application of Electric Welding to Steel Shipbuilding" was presented by Mr. H. A. Hornor, Fellow, A. I. E. E., Head of Electric Welding Branch, Education and Training Section, U. S. Shipbuilding Board, Emergency Fleet Corporation.

The ensuing discussion was participated in by: Mr. M. P. Clark, Mr. Halslag, Mr. C. D. Terry, Mr. A. M. Candy, Mr. A. B. Holcomb, Mr. P. E. Hunter and the author.

On motion the meeting adjourned at 10:16 P. M.

K. F. Treschow, *Secretary.*

BOARD OF DIRECTION

The regular monthly meeting of the Board of Direction of the Engineers Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, December 3, 1918, at 4:15 P. M., President W. E. Snyder presiding, Messrs. Duff, Danforth, Hoerr, Stucki, James, Barbour, Pittman, Hawley and the Secretary being present.

The Minutes of the last regular meeting held November 12 and of the special meeting held 19th were read and approved.

The applications of the following gentlemen, having been regularly published to the Society pursuant to the action of the Board, were elected to membership:

MEMBERS

Landell, John Adams Brunro — Philipsen, Thorulf
Sadler, Jr., Orin Winslow

JUNIOR

Reed, Charles M.

The following applications were received and their names ordered published to the Society. Assignment to the various grades of membership is as follows:

MEMBERS

Carlson, Justus Ewald Nathanae — Jackson, William
Scott, Guy F.

JUNIORS

Lambie, Aaron Louis — Tyler, Lewis Percy

Letters of resignation were presented by the following gentlemen, and after discussion, their names were ordered accepted:

Crea, T. C. — Johnson, George R.
Donahoe, E. E. — Schmitt, A. J.

The report of the Secretary showing the financial condition of the Society October 31, 1918, having been previously audited by the Finance Committee, was approved.

In accordance with the By-Laws, it was moved and carried unanimously that the officers nominated for the year 1919 be approved as published to the Society.

COMMITTEE REPORTS

Mr. Duff, Chairman of the Civic Affairs Committee, reported that no meetings of the committee has been held since their last report and no unfinished business was before the Committee.

In the absence of Mr. Neilson, Chairman of the Entertainment Committee, the Secretary reported an attendance of 120 at the Smoker held November 22nd, and read a statement of the expenditures and receipts.

Mr. Pittman, Chairman of the Finance Committee, reported verbally that the accounts of the Society had been audited October 31, 1918, and were found to be in good shape.

Mr. Barbour, Chairman of the House Committee, reported an evening attendance in the Society Rooms for the month of November as follows:

First week	19
Second week	24
Third week	18
Fourth week	11
First 2 days	4
	76

Mr. James, Chairman of the membership Committee, reported that no meetings of the committee has been held since their last report and no unfinished business was before the committee.

Mr. Danforth, Chairman of the Publication Committee, reported that during the month of November, they held no meetings. The minor matters coming within their jurisdiction, that have arisen, have been such that no meetings have been necessary.

The Secretary stated that Mr. C. E. Oakes, Associate Electrical Engineer, of the Bureau of Standards, Washington, D. C., had called at the office requesting that the Society appoint a committee to represent it on the General Committee on Safety Codes being worked up by the Bureau of Standards at this time; the first meeting of the Committee to be held in Washington on January 15th.

The Secretary was requested to write such members as might be in Washington about this date, asking them to represent the Society.

The meeting adjourned at 5:40 P. M.

K. F. TRESCHOW, *Secretary.*

MECHANICAL SECTION

The regular bi-monthly meeting of the Mechanical Section of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, December 3, 1918, at 8:15 P. M., President W. E. Snyder presiding, in the absence of the Chairman and Vice Chairman.

The Minutes of the last meeting were read and approved.

There being no further business before the Section, the paper of the evening on "The Design of Heating Furnaces From a Practical Standpoint," was presented by George J. Hagan, General Manager, George J. Hagan Co.

The ensuing discussion was participated in by: Mr. Harvey Allen, Eff. Engr., Pressed Steel Car Co., McKees Rocks Pa.; Mr. W. E. Moore, Pres., W. E. Moore & Co., Pittsburgh, Pa.; Mr. H. A. Kunitz, Asst. Chf. Engr., Pittsburgh Crucible Steel Co., Pittsburgh, Pa.; Mr. A. E. Mitchell, Met. Engr., Ordnance Dept., U. S. Steel Corp.; Mr. E. Zademach, Mech. Engr., American Steel & Wire Co., Pittsburgh, Pa.; Mr. F. L. Egan, Engr., River Equipment, Carnegie Steel Co.; Prof. W. Trinks, Prof. Mechanical Engineering, Carnegie Institute of Technology; Mr. R. L. Waggoner, Chf. Steam Engr., Upper Union Mills, Carnegie Steel Co.; Mr. H. E. Gross, A. M. Byers Co., and the author.

On motion the meeting adjourned at 10:30 P. M.

K. F. Treschow, *Secretary.*

REGULAR MONTHLY MEETING

The 373rd regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the Society Rooms, Union Arcade Bldg., Tuesday, December 17, 1918, at 8:15 P. M., President W. E. Snyder presiding, 104 members and visitors being present.

The Minutes of the last regular meeting held November 19th, were read and approved.

The Board of Direction reported the election three applicants to the grade of Member and one to the grade of Junior and the receipt of five applications to membership.

No further business coming before the Society, the paper of the evening on "The Erection of the Ohio River Continuous Truss Bridge at Sciotoville, O.," was presented by Mr. Claude B. Pyle of the McClintic-Marshall Co., Pittsburgh.

The ensuing discussion was participated in by: Mr. N. G. Smith, Designer & Estimator, Ft. Pitt Bridge Wks.; Mr. W. E. Snyder, Mech. Engr., American Steel & Wire Co.; Mr. Paul L. Wolfel, Chf. Engr., McClintic-Marshall Co.; Mr. G. L. Ballou, Cont. Engr., Memphis Steel Construction Co.; Mr. Samuel E. Duff, Cons. Engr.: Mr. George W. Nichols, Engr., S. C. Webb Engineering Co., and the author.

It was moved, seconded and carried that a vote of thanks be extended to Mr Pyle for his very excellent paper.

On motion the meeting adjourned at 10:12 P. M.

K. F. Treschow, *Secretary.*

Proceedings of

The Engineers' Society of Western Pennsylvania

Vol. 34 February, 1918 No. 1

INCORPORATED 1880

OFFICERS FOR 1918

PRESIDENT

W. E. SNYDER

VICE PRESIDENTS

GEORGE H. NEILSON — W. C. HAWLEY

SECRETARY

K. F. TRESCHOW

TREASURER

A. STUCKI

DIRECTORS

HENRY D. JAMES WILLIAM E. MOTT	Term Expires 1919
GEORGE H. BARBOUR E. W. PITTMAN	Term Expires 1920
ROBERT LINTON A. N. DIEHL	Term Expires 1921
SAMUEL E. DUFF ALEX. L. HOERR	Junior Past Presidents
FREDERIC CRABTREE GEORGE H. DANFORTH ALBERT KINGSBURY	Chairmen of Sections

MECHANICAL SECTION

ALBERT KINGSBURY, *Chairman* — L. C. FROHRIEB, *Vice Chairman*

Directors: F. L. EGAN; J. C. HOBBS; C. W. LITTLER

METALLURGICAL AND MINING SECTION

FREDERIC CRABTREE, *Chairman*

Directors: ROBERT LINTON; J. W. PAUL; C. F. TAYLOR; A. W. PATTON; R. R. HICE

CIVIL SECTION

GEORGE H. DANFORTH, *Chairman* — W. M. JUDD, *Vice Chairman*

Directors: KENNETH H. TALBOT; C. W. BRETLAND; C. N. HAGGART

COMMITTEES FOR 1918

ENTERTAINMENT COMMITTEE

GEORGE H. NEILSON, *Chairman*

E. H. COXE
C. M. REPPERT
CHARLES SCHLEY
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